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EDITED BY

THEODORE G. SOARES

THE SUNDAY-SCHOOL BUILDING AND ITS EQUIPMENT

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THE SUNDAY-SCHOOL BUILDING AND ITS EQUIPMENT

By

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GENERAL PREFACE

The progress in religious education in the last few years has been highly encouraging. The subject has attained something of a status as a scientific study, and significant investigative and experimental work has been done. More than that, trained men and women in increasing numbers have been devoting themselves to the endeavor to work out in churches and Sunday schools the practical problems of organization and method.

It would seem that the time has come to present to the large body of workers in the field of religious education some of the results of the studies and practice of those who have attained a measure of educational success. With this end in view the present series of books on "Principles and Methods of Religious Education" has been undertaken.

It is intended that these books, while thoroughly scientific in character, shall be at the same time popular in presentation, so that they may be available to Sunday-school and church workers everywhere. The endeavor is definitely made to take into account the small school with meager equipment, as well as to hold before the larger schools the ideals of equipment and training.

The series is planned to meet as far as possible all the problems that arise in the conduct of the educational work of the church. While the Sunday school, therefore, is considered as the basal organization for this purpose, the wider educational work of the pastor himself and that of the various other church organizations receive due consideration as parts of a unified system of education in morals and religion.

THE EDITOR

FOREWORD

The dominant purpose of this little book is a practical one. One of the great wastes in modern church life is in the construction of the working plants. The rapid adoption of the graded lessons has made an added problem in church construction. This book seeks to accomplish two results: first, to outline, so far as it is possible at the present time, the ideal Sunday-school building; and secondly, to present some of the best recent plans which point toward the degree of efficiency desired in the church school building.

The author wishes to express his obligations to the architects whose names appear beneath the cuts which illustrate the book. Their co-operation makes possible the later chapters. These men will be found reliable and efficient in their profession. The author expresses his appreciation also to Rev. J. W. F. Davies, of Winnetka, Illinois, Rev. Herbert W. Gates, of Rochester, New York, and Rev. Albert W. Palmer, of Oakland, California, who furnished several cuts.

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ILLUSTRATIONS AND PLANS

CHAPTER I

FUNDAMENTAL PRINCIPLES

The guiding principle in this discussion of the Sunday-school building will be *efficiency*. The religious-educational and social needs, especially of the young people of the community, will be regarded as primary. The child is in the midst, not only in the conduct of the worship and the determination of the curriculum of the Sunday school, but also in the construction of the building for his use. This position does not need defense in enlightened circles today. We are beginning to build churches with the fact in view that the child is present in life.

The type of future members in our churches is being determined everywhere in the Sunday schools of today. Let us build so that in the highest degree the facilities are available for the highest efficiency in the realization of our great purpose.

The leisure hours of our young people are potent for good or evil. The church touches the lives of its young people at too few points. The need of direction of young people's leisure time is recognized and the new architecture is responsive to the need.

The Sunday school is regarded as an integral part of the church's activity—its most important service to the community. If this is recognized there will be no objection to the larger amount asked for the provision of adequate buildings and necessary equipment.

The test of *efficiency* does not deny the value of the traditional in architecture. Indeed, properly considered, the beautiful in exterior or interior aids in the high endeavors of teacher or superintendent. These pages, however, will not discuss the traditional forms of church architecture. The underlying purpose is practical, and such matters are left to the architect who breathes the poetry and imagination of past architectural forms.

We turn now to the principles which shall actuate the author in his discussion of the interior of the Sunday-school building. Here the principle of *efficiency* will have full sway. The following principles will be kept in mind throughout the discussion of the problems involved in an effective housing of the church's educational and social activity: (*a*) The importance of beautiful and harmonious arrangement is recognized. The spirit of worship is encouraged by an environment of beauty and harmony. This will lead, for example, to a recommendation to use the church auditorium for the worship period in the Sunday school. (*b*) Although the teaching function of the church

is regarded as of primary importance in this book, the building must be adaptable to other needs of the church as represented by organizations and activities other than those of the Sunday school. Any other attitude than this would be selfish and contrary to the spirit which should dominate the construction of a church building. (*c*) The needs of each department will be determined by investigation and the building will be constructed in such a manner as to respond effectively to these departmental requirements. For convenience the terms adopted by the International Sunday School Association will be used. Those schools which use different departmental divisions will find it possible to make the adjustments without serious difficulty. (*d*) Provision for the individual class will be regarded as of primary importance. An entire chapter will discuss this important matter. (*e*) The efficient building will be related vitally, not only to the religious educational needs of the members of the Sunday school, but also to their physical and social life. (*f*) Facilities for worship will be planned for the whole school, divided into the units demanded by the best results of psychological study. (*g*) The recognized principles of sanitation and hygiene will be regarded as necessary to any correct construction. Religious education is dependent upon good air and light in the accomplishment of its high task. (*h*) Before

proceeding to the determination of the ideal building, we shall profit by the experience of the past. The next chapter will discuss the dominant type of architecture of the last quarter of a century.

CHAPTER II

THE AKRON PLAN

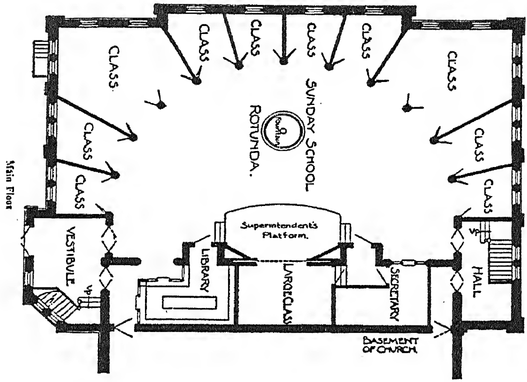
The modern Sunday-school movement started less than a century and a half ago. For over a half-century it was of doubtful respectability. Not until some of the church's far-seeing leaders approved the plan of Sunday schools for religious instruction was the school welcomed into the churches of the land. These early church buildings were of the one-room type with straight pews. Some of the larger buildings had a basement room or two. In the sparsely settled regions of the West, private houses and public-school buildings were used commonly for Sunday-school purposes. The genius of the Sunday school makes it possible to do its important work with little or no facilities or special equipment. The influence of personalities on fire with the ideals of Jesus Christ is not prevented from doing its work by the absence of special buildings. We need at the very outset of our discussion to recognize that ideal buildings and equipment do not *make* a successful Sunday school. The essential element is the consecrated, intelligent teacher. But given a corps of teachers of this type, a good building and equipment greatly multiply efficiency.

With the advance of population and material wealth, and the increase in popularity of the Sunday-school idea, the need of better facilities became pressing.

ADOPTION OF THE AKRON PLAN

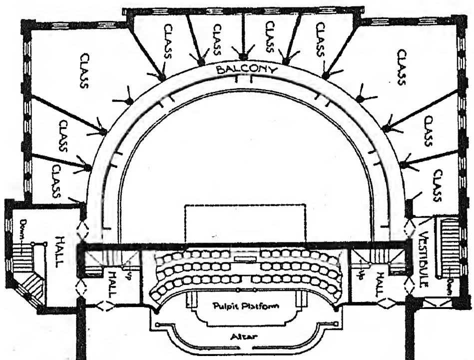
The story of the inception of the Akron plan has been related fully in another volume.[1] Lewis Miller, a lay Sunday-school worker of Akron, Ohio, created the idea which is now known as the "Akron plan." Bishop John H. Vincent defined an ideal Sunday-school room in the following sentence: "Provide for togetherness and separateness; have a room in which the whole school can be brought together in a moment for simultaneous exercises, and with a minimum of movement be divided into classes for uninterrupted classwork." With this definition in mind Mr. Miller, in consultation with Mr. Snyder, a local architect, and Mr. Blythe, a Cleveland architect, prepared the plans for the First Methodist Sunday School of Akron, Ohio. This building was constructed in 1867 and soon became the center of interest for Sunday-school workers and building committees from all over the continent, and indeed from all over the world. The original plan, now familiar to all because of its frequent duplication, is reproduced in Figs. 1 and 2.

[1] Lawrance, *Housing the Sunday School.*



By permission of G. W. Kramer, Architect, New York City

FIG. 1.—The Original Akron Plan. Main Floor



By permission of G. W. Kramer, Architect, New York City

FIG. 2.—The Original Akron Plan. Balcony Floor

It has been stated that three-fourths of the churches during the last fifty years, making special provision for their Sunday schools, have used the Akron plan in some form. The essential features of the original plan were a semicircular auditorium with a balcony; the space under the balcony was divided into classrooms, the front of each being open so that all could see the superintendent's platform; a similar series of classrooms in the balcony provided further facilities for privacy. A row of seats made it possible for all in the balcony to see the superintendent also. A fountain and flowers occupied the center space, which was lighted by clerestory windows. The side walls of these odd-shaped classrooms were plastered and the front was closed by glass doors. Certain obvious advantages were realized at once by this plan. The classes heretofore had not been separated satisfactorily in the one- and two-room church buildings. With the Akron plan many classrooms were made available and much privacy was afforded. The building must have seemed a wonderful advance and quite ideal to the Sunday-school workers of the seventies.

The Uniform Lessons were inaugurated in 1872 and soon the Akron plan showed its adaptability to this system of lessons. All the school studied the same lesson and worship for all was conducted by one man. Perhaps it would be more accurate

to say that "opening and closing exercises" were conducted by the superintendent, who was the all-important individual. Most of our Sunday schools fail to use effectively the time for worship. Plenty of noise and "enthusiasm" led by a popular business man is mistaken often for successful worship in the Sunday school. The Akron plan lent itself to the "togetherness" idea of Bishop Vincent. One of the features of the Uniform Lesson system in later years has been the superintendent's five-minute review of the Uniform Lesson at the close of the class-study. This was facilitated by the Akron plan. For forty years most of the churches have used some form of the Akron plan in their provision for Sunday-school instruction. The plan was varied in many ways, such as the development of a more satisfactory balcony seating arrangement, the introduction of more effective means of shutting out sound, the squaring of the classrooms, and the seating of the center space on the main floor. But the essential idea of "togetherness" has reigned supreme.

FAULTS OF THE AKRON PLAN

With wide experience in the use of buildings of this type difficulties arose. Discriminating Sunday-school workers discovered that these difficulties were largely due to the way in which the building was constructed. The many partitions

of the Akron plan created problems of discipline that interfered seriously in the efficient conduct of worship. Thoughtful workers realize that worship is one of the great opportunities of the Sunday-school hour. Any lack of efficiency here must be studied carefully and removed. The worship of God is a social act and needs to be conducted in a spirit of fellowship. The numerous plastered walls of the Akron plan broke the Sunday-school congregation into segments and prevented the helpful worship possible in the open room of churchly architecture. Architects attempted to obviate this difficulty by providing balcony seats from which a portion, at least, of the Sunday school could be seen; but this was not wholly satisfactory. One of the best known of our Sunday schools has not for years allowed classes to go to the balcony, except for the class hour. This, no doubt, is but one instance which might be multiplied by hundreds.

We are still controlled largely in our Sunday schools by the "togetherness" idea. The Akron plan will be considered of great advantage as long as the superintendent is regarded as the most important personage in the school. Just as long as he is considered the most efficient individual to conduct worship for *all*, just as long as the superintendent's review of the lesson is considered as superior to the specialized summaries of the indi-

vidual teachers, the Akron plan will lend itself to the "togetherness" idea.

As better methods of instructing young children came into our Sunday schools the teachers of Beginners and Primary departments began to demand entire separation. This has been realized only gradually and with great reluctance on the part of the Sunday-school world. But the inevitable has come to pass and reflects itself in separate rooms for the two lower departments. George W. Kramer, a New York architect, apparently saw the trend of Sunday-school development and as far back as 1893 prepared a model building which was exhibited at the World's Fair that year. This building provided separate rooms for all departments, still retaining, however, the possibility of "togetherness."

Nor was the method by which "separateness" was gained altogether satisfactory. Reference has already been made to the demands of the Primary teachers. The requirements of pedagogy call for frequent change of program with the little children. The use of music could not but disturb the rest of the school. The strangely shaped classrooms required for the focus of all upon the superintendent were not satisfactory. Frequently they were poorly lighted, and almost always, especially in the smaller churches, the ventilation was seriously defective. So often is this the case that

very frequently the curtains or doors of the Akron-plan classrooms are left open during the class hour. Poor air and light will defeat the best efforts of an efficient teacher. The elimination of disturbing noises was not completely successful, because of the flimsy partitions and temporary provision for the front closing so frequently used.

THE AKRON PLAN AND GRADED LESSONS

In 1908 the International Sunday School Association in convention at Louisville, Kentucky, authorized the preparation of graded lesson outlines. The graded lessons for the lower departments had been in use for several years. The architectural situation soon became acute. Within five years 35,000 schools adopted the graded lessons. These schools felt the pressure of inadequate building facilities. The Akron plan was found to be unsuited to the new lessons. This was foreseen by the more intelligent of the advocates of the Akron plan. The graded lessons require several separate departmental assemblies as well as separation by grades or classes within departments. Each grade in a fully graded school uses different lesson material. The review by the superintendent is no longer possible. The superintendent of a school of 1,000 members told the author that he was going to resign, for there was nothing worth while for him to do, now that the

lesson did not require review at the close of the hour! With the new graded lessons a general assembly is only occasional, except in the smaller schools. Stress is laid upon a carefully conducted departmental assembly, and especially upon the work of the pupils in the individual classes. The new unit is the classroom. The new person of importance is the teacher.

There has been much activity among architects since 1909, seeking to produce a building which shall be satisfactory with the new graded lessons. Much progress has been made. This volume will show plans of some of the newer types and will indicate the line of progress. But before considering the plans in detail it will be advisable to determine the lines which an ideal Sunday-school building will follow. This we shall do in the following five chapters.

CHAPTER III

THE IMPORTANCE OF THE EXTERIOR

THE IDEAL SUNDAY-SCHOOL BUILDING

It will, of course, be impossible to present in the following chapters the plans for a building which will satisfy every need of every Sunday school, regardless of situation and size. However, there are certain fundamental features which should be present in the ideal building for the modern graded Sunday school. Indeed, those churches which still prefer to use the Uniform Lessons will find here helpful suggestions in planning their new buildings. The question is asked in every case, Will this feature make the building more efficient in carrying out the ultimate purpose of the Sunday school as the educational arm of the church? The author has in mind the average school rather than the very large or the very small school. Practically every suggestion will be found adaptable in some measure to every school. The reader will find in the succeeding chapters the principles and illustrations by which he may work out the solution of his individual problems.

EXTERIOR ARCHITECTURE

The purpose for which the Sunday school exists can best be served by a substantial, dignified, and

beautiful exterior. If beautiful architecture can be justified at all, it must be used in the buildings which house the religious educational facilities for our growing young people. The church building should incorporate in itself, in a sense, some of the great thoughts for which religion stands. It is a reflection of the value which its builders place upon God and his worship.

Therefore it should be durable in construction, with simple exterior plan and notable absence of flimsy ornament. That construction material which is genuine, rather than that which is veneer or showy in character, is to be preferred. The lines of the building should suggest strength and repose, dignity and reverence. Thus the unconscious impression of the building in which the Sunday-school interests of the church are housed will serve that for which the Sunday school exists. It will take a courageous committee to withstand the temptation to make a large, showy exterior. The ministry of art in giving refinement and proportion to our church buildings is an undoubted influence for religious education. While these considerations are usually given due attention in our larger and more pretentious buildings, they are often forgotten in our humbler churches, which, nevertheless, are planned to perform the same function in the lives of our people.

Attention should be called also to the necessity of surrounding our churches with artistic and well-kept grounds. Many otherwise beautiful buildings give a poor impression because of the wrong treatment of the grounds. The buildings are oftentimes placed too near the street, or face the street in a wrong direction. The vertical surface of the side of the building should be blended with the horizontal surface of the ground by judicious planting of shrubbery or vines. The landscape gardener often can render great service to the committee just completing a new church and Sunday-school building. Too much attention cannot be given to the impression made upon boys and girls by the buildings and grounds with which is associated their religious education.

The buildings of the Second Baptist Church, St. Louis, Missouri (Fig. 3), illustrate in a marked degree the use of the splendid Lombardy Gothic architecture. The left-hand building is used for church worship and the right-hand building for educational and social purposes. The space between is a sunken garden. Above the rear loggia rises the noble campanile. Thus the church in its architecture declares to the world its faith in the importance both of worship and the educational and social work of the church.

An excellent example of a good exterior for a church seeking to serve the community at large is

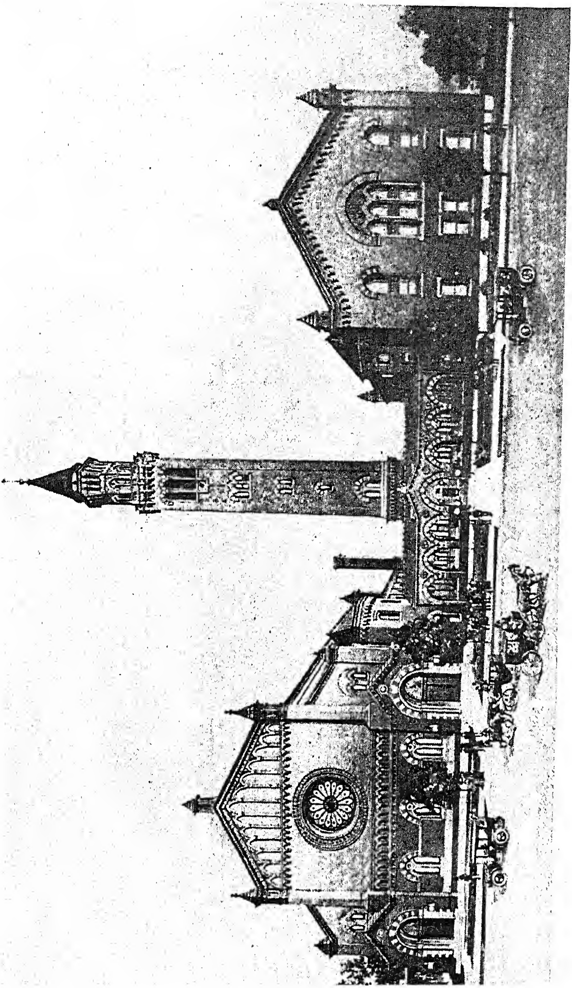
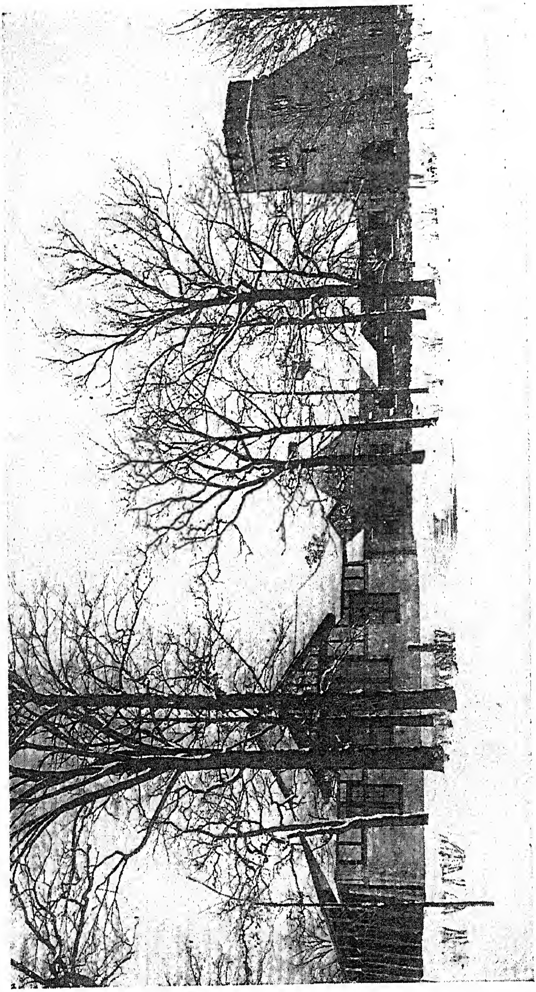


FIG. 3.—Second Baptist Church, St. Louis, Missouri



Courtesy of Rev. J. W. F. Davies, Winnetka, Ill.

FIG. 4.—Winnetka (Ill.) Congregational Church

shown in Fig. 4. This church is located in a suburban district where the cost of the land is not so serious a factor as in a large city. The plans of the community house of the Winnetka (Ill.) Congregational Church are shown in Figs. 17, 18. The notable absence of the showy, the noble tower entrance, the blending of lawn and shrubbery with the rough stone of the walls, all contribute to a harmonious whole which must have a refining influence upon the hundreds who come and go every week.

It is not necessary to expend a large amount of money to gain a building the exterior of which is satisfying. Often a large amount of money fails to gain the very elements which are here regarded as essential. It is more a matter of good taste and artistic judgment than the expenditure of money. If the architect employed is an artist he will breathe into the church, whether large or small, an atmosphere of reverence and worship.

CHAPTER IV

SUNDAY-SCHOOL WORSHIP AND GENERAL ASSEMBLY

The worship of the pupils of our Sunday schools should be graded as well as the curriculum material. This is made necessary, not by any passing fad, but by the newly discovered growing child, who determines for us the proper equipment for his education. It is not possible to conduct properly in one session the worship of pupils varying in age from four years to full maturity. When the appeal is effective for one age, other groups are listless or inattentive. Thus the problem of correctly grading Sunday-school worship becomes a vital one to the persons seeking to house the Sunday school effectively. An understanding of its function in directing the worship of children and youth will urge careful attention to that portion of the Sunday-school activity which takes one-half of the pupil's time in the Sunday-school session. Psychology, then, demands that the Sunday-school worship shall be graded and conducted in several groups.

When the school is large the problem is not so difficult. The Beginners will have a separate room with musical instrument. So also will the

Primary children, aged six to eight. Some schools find it possible to conduct worship in an "Elementary Division," constituting Grades 1 to 4 or 5 (Fig. 6). The Juniors in a large school will worship in their own room, effectually shut away from outside disturbances. This department should be cared for generously in every way, for the church's recruits are largely prepared for the new loyalty in this period. Worship, then, should be directed carefully for the Junior group. The Intermediate and Senior departments will worship together most frequently. It is true that a demand for sex separation in the "teen-age" periods calls for two groups for worship. Where this seems to be wise, separate rooms will be required. Later plans will show how this is possible. The great majority of schools, no doubt, will continue for some time the worship of ages thirteen to twenty of both sexes in one group. Separate assemblies of boys or girls after class work can be planned easily. Adult worship comes normally at the church hour. No special provision is needed for men and women in most schools. Either they will worship with the upper departments of the school, or, meeting a half-hour later than the rest of the school, they will go into the regular morning service for their worship.

When the school is smaller, combinations will be necessary and desirable in worship. The Beginners

and Primary departments will unite for worship, and the remainder of the school will worship together. Such a plan will be shown later (Fig. 10).

But what provision will be made in the ideal building for general assembly of the whole school? There are the special days with large attendance and many visitors—Rally Day, Thanksgiving Day, Christmas, Easter, Memorial Day, Graduation, etc. The whole school will meet in general assembly only occasionally, not over seven or eight times a year. A large auditorium should not be built to be used on so few occasions. Where shall these sessions be held? The obvious answer is, In the church auditorium. Some may object at once. The church proper should not be used for children's exercises lest reverence be destroyed. One of the important duties of the church school is to develop a sense of reverence in the growing child. Surely no place could be found more calculated to arouse reverence than the church auditorium. It is true that the use of the church auditorium might change the type of worship and general assembly. That might not be a serious result. We suspect that the unwillingness to allow the use of the church auditorium for Sunday-school purposes, in many cases, arises from a knowledge of the type of worship often conducted in our Sunday schools.

In the larger churches there will be a demand for a general assembly room for entertainments

and social purposes. Such a room should then be provided in the ideal plan. Often it will be the gymnasium where facilities are afforded for physical recreation (Figs. 8, 13, 39 will illustrate this suggestion).

The worship facilities of departmental groups will be discussed more fully in the next chapter. This chapter has sought to make clear the fact that the worship of a modern Sunday school will be conducted in from three to five groups, according to the size of the school. The secondary auditorium for Sunday-school assembly has been discouraged, for the space is needed for classrooms, as chap. vi will make clear. Where the Sunday school is large, a secondary auditorium, also fitted to serve the church for social and gymnastic purposes, is recommended. The plans shown later will illustrate the possibilities in this direction.

CHAPTER V

MEETING THE NEEDS OF THE DEPARTMENTS

Departmental needs will be discussed under the following heads: Beginners, Primary, Junior, Intermediate, Senior, Adult. Access to any of these departments must be direct from halls and not through other departments. The classroom facilities for each department will be discussed in detail in the following section.

It is clear that absolutely separate rooms must be provided for the first three departments. The method of combining these departments into a large assembly room by the use of temporary partitions of any kind must be abandoned, and perfect freedom from disturbance by those near by must be accorded each of these departments. Each of these three departments ministers to a distinct epoch in the life of the child or youth. If a given department does not do its full work, all later departments will suffer in their efficiency; or, stated in more serious language, the religious life of the boys and girls in the departments not properly provided for will suffer beyond recovery.

BEGINNERS DEPARTMENT

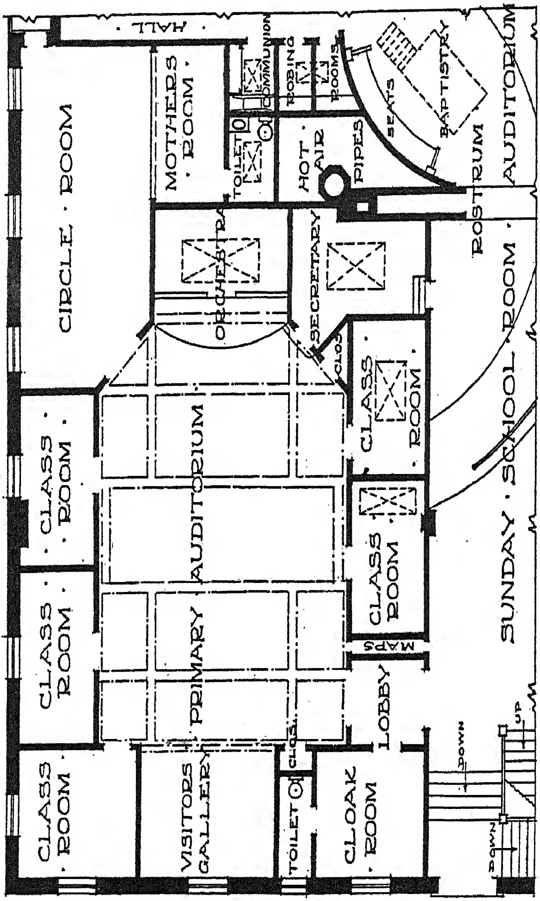
No movement in modern education has better vindicated its right to be than the Kindergarten.

The religious significance of the work for the smaller children is well recognized by educators. The Beginners Department in the Sunday school, then, should have every facility for its work. It would be foolish policy indeed to limit the efficiency of the educational work at its very foundation. How permanent can we expect the superstructure to be that is placed upon an inadequate foundation? The social experience of the children of the Beginners' age is limited very largely to the home. The room used for their religious education should therefore partake as largely as possible of ideal, homelike conditions. The ideal Beginners' room will be flooded with sunshine and cheeriness and provided with ample fresh air. The young child in a new environment will be fearful if the place is gloomy. The department will be on the ground floor with the fewest possible steps. Even two or three steps should be eliminated, when direct outside entrance is possible, by the use of a rubber-covered gentle incline. The ceiling of the ideal department for Beginners will be low and studded. Care will be taken that the room is not unduly large. The department needs little more room than for the circle of chairs and the kindergarten tables. The visitors should have an inconspicuous place at the backs of the children; possibly, if the school is large and visitors are present often, in an alcove built a step above the room. The pictures

used to decorate this room should be hung low, near to the line of vision of the children; a burlap dado is useful on which to fasten lesson pictures close to their eyes. The room will be more homelike if the floor is covered with a rug. If bare floors are used, the legs of the little chairs should be provided with rubber tips. The blackboard is desirable, either built into place or movable. The children can do their work best seated on small chairs at standard kindergarten tables. Thoughtfulness for the teachers dictates ample locker and cabinet space, so that all lesson material may be stored in order and be found quickly when needed. The lack of this simple requirement often has interfered with efficient work. The ideal calls for a cloakroom; a closet with low toilet for children; and a screen near the entrance of the room to prevent undue disturbance from those entering during the exercises. Screens will also be found valuable for the separation of the two years of the Beginners, and of classes seated close to each other. In the larger school a folding door, or other device, may be helpful during the class hour for the separation of the two years of Beginners' work. Other suggestions for this department may be found in the examination of modern kindergarten departments in our better public schools.

PRIMARY DEPARTMENT

This department should have a room entirely its own, entirely separated from other departments by permanent walls. It should have access to the rest of the school by means of halls, not by means of doors entering directly into other departments. Where the Primary and Beginners departments are located in close contiguity, a cloakroom may, with advantage, be placed between the two rooms. The Mothers' room, to which reference is made later, may be located with convenient access to these departments. The Primary Department room should be large enough to permit of division into three rooms by means of accordion doors or other device. This will permit each grade to have a room by itself for the class hour. Should this not be practicable, a compromise may be made by having two smaller classrooms opening from the main Primary room. All that has been said in the paragraph on the Beginners Department concerning sunshine, height of ceiling, placing of pictures, tables, and chairs is applicable here. The tables should be of a height that will enable the pupils to work with comfort. The chairs should enable the children to sit with their feet easily reaching the floor. Fig. 5 gives the floor plan of the Long Beach (California) Methodist Primary Department which has many admirable features. Note the provision for children's toilet, visitors, separate



Norman E. Marsh, Architect, Los Angeles, Cal.

FIG. 5.—Long Beach (Cal.) Methodist Church. Primary Department

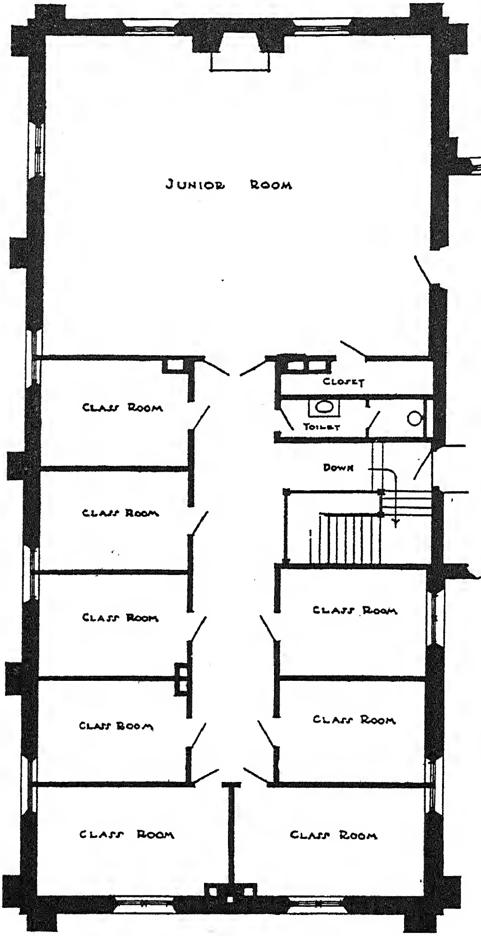
classrooms, and soft, overhead light. The Plymouth Church, Minneapolis, has a department for Grades 1 to 4 (Fig. 6) which is thus described by the superintendent of the school in *Religious Education* (August, 1910):

> The Junior Department includes the Kindergarten class and the first four grades, and for this section of the school the Junior or Children's room was designed. Accessible through a large double door, it is a room 30×34 feet in size. In one corner is a door which leads to a safe iron fire escape. The room is lighted by eight Gothic windows. The woodwork is a soft brown-toned oak, the walls painted in flat color to harmonize with the panelings. A good yellow-brown carpet covers the floor; simple net curtains soften the light which comes through the many diamond panes of clear glass. The room is furnished with 120 specially designed little Gothic chairs in the same soft brown color.

A unique feature of this room is the generous use of the best art in its decoration. A beautiful fireplace is centrally located. On the wainscoting of three sides of the room are installed forty-four brown carbon prints of the life of Jesus. Opening from this room are enough classrooms to allow each grade to withdraw to its own room, leaving the larger room for the Beginners' circle.

JUNIOR DEPARTMENT

Some of the most important work in the Sunday school is done during the four years of this department. More study may be expected and more



Shepley, Rutan & Coolidge, Architects, Boston, Mass.

FIG. 6.—Plymouth Church, Minneapolis, Minn. Junior Room

information is absorbed by the pupils during this period than in any other to which the Sunday school ministers. A separate departmental room is absolutely essential in which worship can be conducted without disturbing other departments or being disturbed by them. The same suggestions made earlier concerning cloakrooms may be used in this case to insure soundproof partitions.

The room should be capable of division into four separate grade rooms by removable partitions. Experts vary as to the separation of the sexes for class work in this department. The writer regards the separation of boys and girls as desirable at this age. In such case the provision of four additional classrooms opening from this departmental room would be ideal. Where the divisions are made as first suggested, screens may separate the classes in the same room. These classes will be seated at tables about 3×7 feet in size, the teacher seated at the middle of one side of the table. Where provision is made for a geography room it should be located in convenient relation to the higher grades of the Junior Department. See a later paragraph for a description of this room. Blackboards should be available for each class in this department, and maps for the upper classes, depending on the grade in which they take up geography in the public schools.

The Junior Department is a busy workroom, having no special provision for visitors. This department needs every facility for worship and for grade instruction, and in the larger schools for separation into individual classes not exceeding ten pupils in number.

INTERMEDIATE DEPARTMENT

The architectural requirements for the Intermediate and Senior departments vary with the size of the school. With the average school these departments will probably meet together for worship, also including adult members of the school. In this case there will be required a room of adequate size for the assembly with a sufficient number of adjacent classrooms of varying size. The assembly room may also be divided into several temporary classrooms. Not every grade of the International system will be represented always in these departments. It will be better to group a larger number with a fine teacher than to break up these departments into numerous small classes which will lack the essential quality of enthusiasm.

There is developing a considerable sentiment for Boys' and Girls' departments from the Intermediate age on. Where this is desired, adequate architectural provision can be made in a manner similar to that recommended for the Junior Depart-

ment. Illustrations of such a division will be shown later (see Figs. 19, 20, 35). In the larger schools the Boys' and Girls' departments will be found of considerable advantage, especially if provision is made for regular worship together. The larger school will make provision for separate assembly for the Intermediate and Senior departments. The assembly room of the Intermediate Department when provided could be divided into two rooms, one for each sex, for departmental meetings. Close to this assembly room the classrooms should be located.

The use of the church auditorium for the worship of the Intermediate, Senior, and Adult departments is recommended where the school is not too large. This would give a beautiful, churchly room for the worship of these groups and would obviate the necessity of building a second auditorium for this special purpose. A later section will indicate more in detail the character of the classrooms.

SENIOR DEPARTMENT

The needs of this department have been covered practically in the foregoing section. The unit is the classroom of the type suggested in the discussion of that important subject. The classes will tend to become larger in size in this department, hence larger classrooms for organized classes will be required.

ADULT DEPARTMENT

The worship provision for the members of this department should be either in the united session of the upper departments referred to in foregoing sections or in the regular service of worship of the church, which is the logical time for adult members of the school. In that case adult classes will meet in their own classrooms a half-hour later than the rest of the school if the session precedes the morning worship of the church. Large, cheerful rooms, comfortably seated, provided with built-in black-boards and a nest of maps of biblical lands, will provide adequate accommodation for adult classes. These rooms may be thrown together by means of folding doors and thus make the large church parlor for general social occasions.

CHAPTER VI

THE CLASSROOM—THE NEW UNIT OF CONSTRUCTION

The classroom is the unit of architecture for the graded lessons. The teacher is the important personage whose class of whatever age must be given adequate provision. In general an *ideal* classroom may be described as a rectangular, plastered room, with outdoor light and good air. This room will have entrance by but one door to a hall, and will not be connected with a neighboring classroom, except by this hall. Wall space will be sufficient for all equipment, including maps and blackboard necessary for the conduct of the class. A cabinet will be in place in which the class supplies can be kept. The floor space will be sufficient for a large table about which the class will sit, or in the case of high-school pupils desk chairs may be substituted. Upon the walls will be hung beautiful art reproductions suitable to the age using the room and appropriate to the lesson material studied. Clearly this is an ideal situation which in many cases must be approximated rather than fully realized. But it is well to recognize the ideal; often it will be found quite possible of realization.

CLASSROOMS BY DEPARTMENTS

In the Beginners and Primary departments separate rooms for a portion of the membership are desirable in some degree, but not so essential as in later years. Screens, curtains, and folding doors will frequently afford such a degree of privacy and freedom from disturbance as will give efficient service. The larger the departments the more provision should be made for some additional classrooms for these departments, but in the average school such provisions as are suggested in the previous section may be regarded as adequate.

The Junior Department, however, presents a different problem. Discipline must take a different form. Outside interruptions must be shut out in every way possible. The author regards separate classrooms as pedagogically valuable for this department, or for the upper classes in it. At least screens or curtains should be employed. If these Junior classes can be shut away from outside noises and sights, efficiency will be greatly increased. A much larger number may be handled in a class when a quiet room is provided. The classrooms for the Junior Department should have large tables of proper height, comfortable chairs, blackboard, suitable pictures, and, in the upper grades, maps of Palestine.

It is in the Intermediate Department that the classroom is of the utmost importance. Answering to the general requirements of the ideal classroom it may also become the clubroom for the social life of the class during the week. Its decoration may be made a matter of class interest under the direction of the teacher. Knowing that 60 per cent of all the pupils who leave the Sunday school do so during the ages which this department includes, what should we not do to make the church life of these unstable youths of the utmost attractiveness?

All that has been said concerning the classroom requirements of the Intermediate is true also of the Senior Department. These young people will very soon be active in many of the church organizations. Let them have every encouragement. The church should be the most attractive place in the community life to them. Churches which desire to improve present buildings by providing better classroom facilities will find suggestions in chap. xii, "Remodeling Old Church Buildings."

A word may be added about different methods of making classrooms. Curtains are better than nothing, but should not be planned in a new building. In one of the recent notable Sunday-school buildings from the standpoint of expenditure, a sum of $1,200 was expended for curtains and brass

rods to make sixteen classrooms! A few hundred dollars more would have given a much superior form of separation of classes. Screens are good to separate classes from passing people, but are not efficient in shutting out noises. Accordion doors, when tightly fitted, or flexible doors similar to a roll-top desk are good. Architects are using a door consisting of a frame covered with heavy canvas on either side and inclosing an air space. This door or partition is said to be very effective. The architect should be consulted about these details. Nothing, however, will take the place of the plastered wall and the closely fitted door.

SPECIAL ROOMS

There are a number of special rooms, several of which should have place in every progressive Sunday-school building. The director or superintendent should have an office conveniently situated with reference to the activities of the school and easily accessible to the public on week days, especially when the director is a salaried official and keeps regular hours. The teachers should have a room to which they may come at any time for study. It should have facilities for keeping books; should have a comfortable table, and good light for reading. It should be large enough for the weekly or monthly meetings of the teachers. It might profitably be *en suite* with the museum, missionary,

and exhibit room, and the geography room to which reference is now made. The museum, missionary, and exhibit room serves a threefold purpose, for within its walls should be brought together every object that will help to illuminate the Bible, which is essentially an oriental book, objects which will help the pupils of the school to understand the activities of missionaries, and lastly an exhibit of the work of the pupils of the various grades. The knowledge that their work, if of sufficiently good quality, may be exhibited will be a legitimate incentive to many. The geography room is in line with the tendency in our best schools toward departmental methods in teaching a difficult subject. This room will be equipped generously with the best maps, topographical maps, globe, sand-trays, work-table, etc., and will be in charge of an expert teacher in geography. Classes of various grades, especially those of the late Junior and early Intermediate ages, will receive in this room the special geographical instruction which will enable them to pursue their regular courses intelligently. The secretary and librarian should have good rooms with convenient facilities and ample cabinet space for supplies. In the largest schools all of the extra supply equipment may well be kept behind a counter, which will enable the secretary and librarian to meet all needs in an orderly manner. The Mothers' room,

situated close to the Beginners and Primary departments, has been found to serve a good purpose. This room can be made of additional value by equipping it as a classroom in child life for the mothers who wish to be near their children.

CHAPTER VII

THE CHURCH-HOUSE AND COMMUNITY SERVICE

One of the most serious problems of modern life is the proper use of leisure time. Many of the evils of city life are caused by misdirected activities of the hours when labor or school does not keep young people occupied. Boys and girls and young people are on the streets and become patrons of commercialized amusements, most of which are superficial and give distorted views of life. The motion-picture show, as now conducted, is frequently positively harmful and imparts conceptions of life at serious variance with those ideals for which the church stands. The stress and struggle for physical stability in young people requires abundant expression of the play impulse. In our modern life there is scant opportunity for this expression. The national game is splendid for the players, but the effect is not permanently good for the thousands of young people who crowd the bleachers. The influence of the amusement parks in our large cities is almost uniformly harmful. The student of conditions discovers a generation of young people who are giving their leisure time to a search

for pleasure which often proves to be pleasure of an unwholesome type. This search takes them far from the church, the doors of which are frequently closed during six days of the week.

There is developing rapidly in our churches a desire to serve the community at the point of need, whatever that may be. Developing the kingdom of God on earth is becoming a dominant motive rather than the preparation of people for life in another world. In the new social era upon which we are just entering, some organization should become the educative influence toward higher civic ideals. Why should not the church accomplish this task for the community? Who shall provide a center for social and recreational activity? Why not open the church buildings and put them to use seven days in the week instead of only occasionally, as in so many cases now? The great play impulse deeply implanted in human nature should not be wholly commercialized. The aesthetic nature of man should not be turned over entirely to grand opera.

This chapter is concerned with the architectural needs of the church which is seeking the largest possible service to the community of which it is a part. Obviously these needs will vary widely and the demands upon some churches will be much greater than upon others. As long as boys and girls need recreation and social life the church is

charged with the duty of either supplying the opportunities or being sure that other community agencies provide wholesome recreation. There should not be serious duplication of agencies. If a Y.M.C.A. provides *ample* opportunity for physical exercise and play it would be better for the church to support the agency already established. It will be better to censor strictly the motion pictures of a dozen theaters, if the proprietors will submit to a censorship, than to provide the pictures for a comparatively small number in the church-house.

A large social hall and gymnasium combined will be found of great service to the community. Here entertainments and musical occasions can serve the community at large. The play instinct of young people can find full opportunity for expression. This room should have a high ceiling and should usually have a gallery for spectators. The room should not be stocked with a full assortment of apparatus. This is expensive and unnecessary. Better results may be obtained with games which excite the interest of the boys and girls. The room should be as large as possible in floor space, up to 50×80 feet in size. A stage at one end will give opportunity for amateur dramatics. At the other end a fireproof motion-picture operating-room can be constructed. Folding chairs should be provided for audience purposes. Special

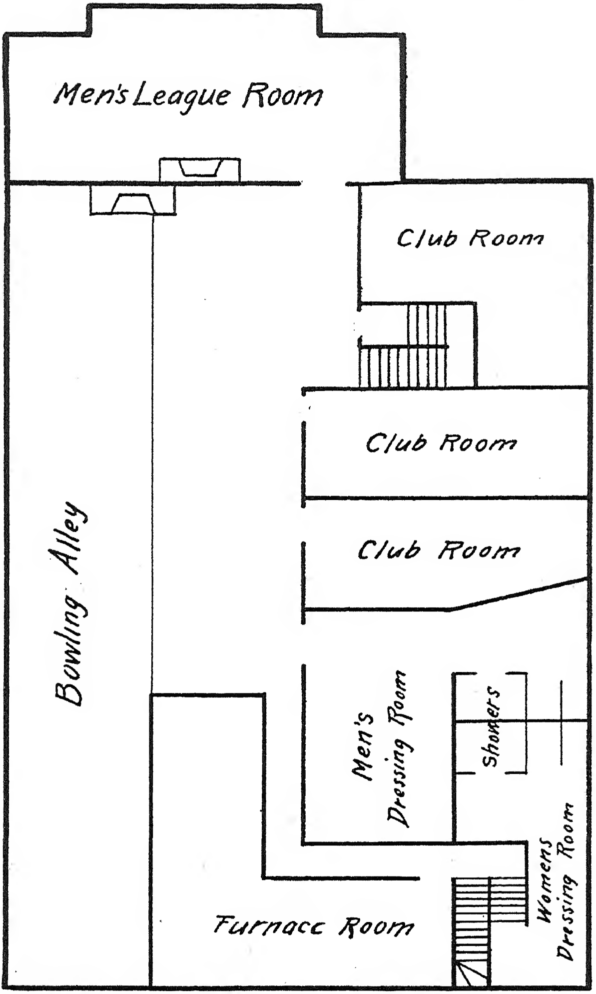
attention should be paid to exits from this large general room. They should be ample in size, not less than two in number, preferably four if it is a full-size room, and stairs should be as few as possible. Single steps in halls are an abomination and should be avoided absolutely. Inclines are preferable, especially when large numbers of people use the halls for entrance and exit purposes.

Where the opportunity for exercise is afforded it is essential that lockers and shower baths be provided also. And in this day of the "boy problem" we must not forget to provide for the girls also.

The church-house seeking the largest service to the community will provide, in addition to entertainment and gymnasium features such as have been outlined above, rooms for reading and games, in case such service is not rendered adequately by the community library or other agency. It cannot be concluded that this service is unnecessary because there is a public library or a Y.M.C.A. a mile or two away. Perhaps a branch ought to be located in the church, which many public libraries are very willing to permit. The direction of children and youth in their reading is one of the most potent, and as yet neglected, opportunities for character development. Rooms should be provided for the club life of the boys and girls. The clubs under skilled direction are powerful

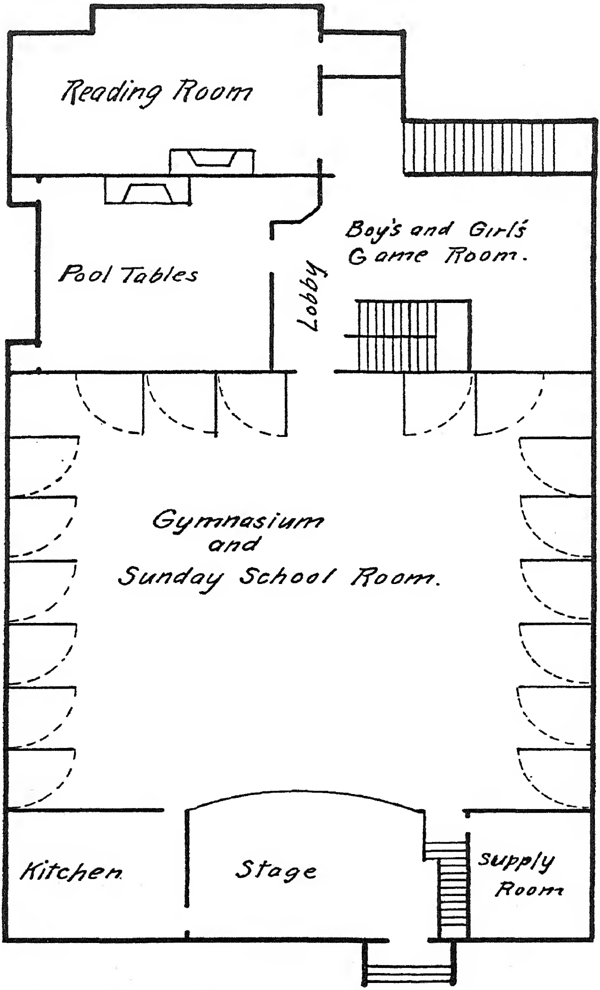
factors in the creation of high ideals. These same rooms are, of course, available as classrooms for the Sunday school. Attractive parlors for social life, with convenient kitchen arrangements near by, are also valuable for the largest community service. Special community needs often make it advisable to have provision for billiards and bowling in the community building. Provision for bowling should be in the basement and should be so located that the noise will not disturb other activities. Let those who would criticize sharply the provision for billiards and bowling in the church-house ask the question of themselves, Are buildings more "sacred" than boys?

There are scores of church buildings today which approximate the service to the community described above. Plans of the Winnetka (Illinois) Congregational Church and the St. Paul's Methodist Episcopal Church, Cedar Rapids, Iowa, shown in later chapters, are especially good illustrations. Note the plans offered in all of the later chapters. There are shown here (Figs. 7, 8, 9) the three floors of Plymouth Center Building, Oakland, California. In a very few years this church has multiplied its membership fivefold and crowded its two buildings by rendering the type of service indicated above. The plans largely explain themselves. In the basement there is provision for bowling, a men's league room, several clubrooms



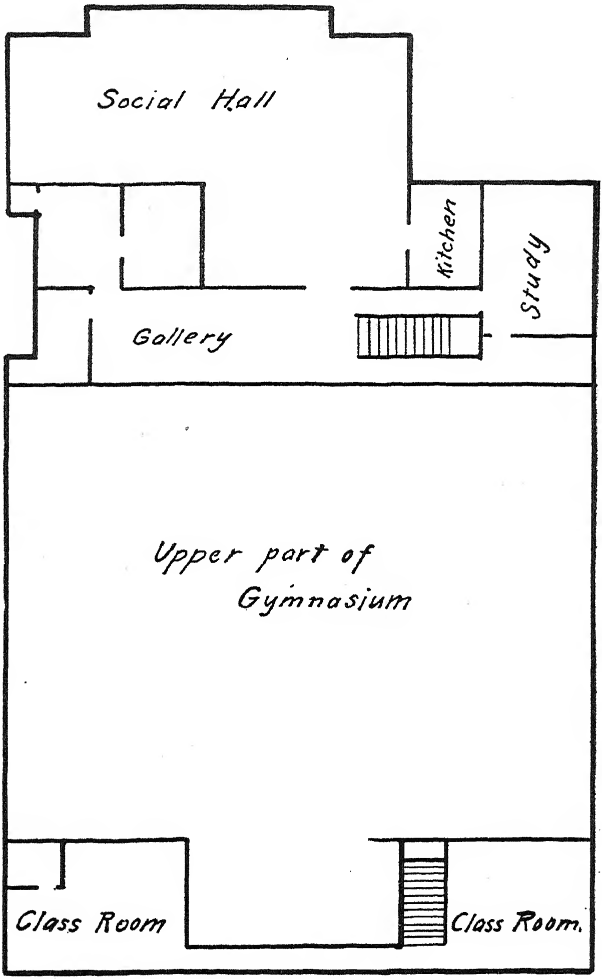
Courtesy of Rev. A. W. Palmer, Oakland, Cal.

FIG. 7.—Plymouth Center, Oakland, Cal. Basement



Courtesy of Rev. A. W. Palmer, Oakland, Cal.

FIG. 8.—Plymouth Center, Oakland, Cal. Main Floor



Courtesy of Rev. A. W. Palmer, Oakland, Cal.

FIG. 9.—Plymouth Center, Oakland, Cal. Second Floor

for boys, and showers and dressing-rooms. The first floor provides a boys' and girls' game-room, a reading-room, and accommodations for pool tables. The use of this building is scheduled so that girls and young women are in exclusive possession at stated periods. A large gymnasium and Sunday-school room occupies most of the first floor. Alcove classrooms are provided by large doors which swing out from the side walls. During the week these make a wainscoting for the gymnasium. This is not an ideal arrangement for classrooms, as an earlier chapter has indicated, but at least eye-disturbances are avoided. This room, 55×65 feet in size, has a 22-foot ceiling, thus making an ideal play gymnasium. The second-floor plan shows a gallery for spectators, classrooms, a study for the pastor, and a social hall with adjacent kitchen. This plant is separate from the church building proper, and cost $25,000. The response of the community has been such that already it is overcrowded.

While providing for the week-day activities of the church this building lends itself well to the Sunday-school class work. Every portion of the building is occupied with Sunday-school departments and classes. In a short time the Sunday school has grown to about six hundred members. The Junior Department occupies the gymnasium; the Girls' High-School Department,

the reading-room; the Boys' High-School Department, the men's league room and adjacent classrooms; the Primary Department, the social hall.

Our churches are increasingly providing for this larger service to the community. It will be well for all building committees to consider carefully the rich opportunities in this direction.

CHAPTER VIII

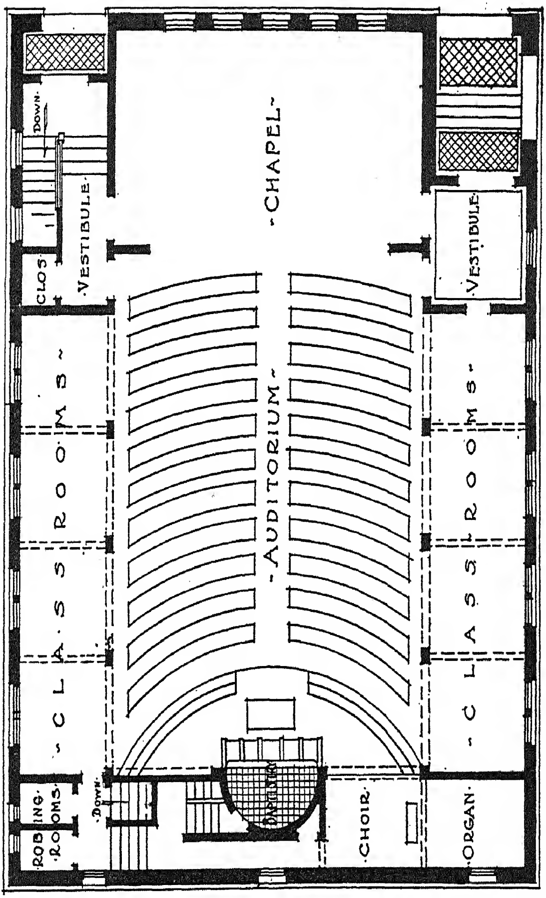
THE VILLAGE AND COUNTRY SUNDAY-SCHOOL BUILDING

Much is being written on the rural problem at the present moment. There is not space here to discuss all of the problems which have been shown to be present in our village and rural situation. When the investigations have been completed it will be found, no doubt, that men and women, boys and girls are essentially the same, whether they live in the crowded city or in the small village or in the open country. Their desire to worship, their need of knowledge of the Bible, their need of social and recreational activity will be found to be similar to that of their city cousins. The country church will seek to develop its possibilities to supply that which the community needs for its larger life. At the present time the average village and country church building consists of one or two rooms. Chap. xii will suggest some ways by which these buildings may be improved. This chapter seeks to give some suggestions to the village or country church which is about to construct a new building.

WESTERN (NEBRASKA) BAPTIST CHURCH

The ideals suggested in the preceding chapters are not entirely impossible for the village or country

church. Not everything mentioned is required for the smaller group of people to which such a church ministers. Fig. 10 gives the floor plans of the Western (Nebraska) Baptist Church, altered by the courtesy of the architect to approximate the needs of the graded lessons. An examination of this plan will show that separate departmental sessions will be possible for Beginners and Primary, Junior, Intermediate, Senior and Adult. For a school of approximately one hundred and fifty pupils separate classrooms will be provided for all of the classes of the Junior and Intermediate departments on the basis of combining two grades in one class in each case. This is more desirable than attempting a full-graded plan with only four or five at most in a class. The auditorium is lighted from above when classrooms at the sides are closed. The Primary Department would have an excellent room in the chapel. Senior and Adult classes could meet in the corners of the auditorium, which could be used for the worship service of the whole school from the Junior Department up. The chapel room would be found excellent for the social life. The number of classrooms could be doubled by building a second story of them on each side with stairways from the vestibules. A basement floor would give added facilities for social and recreational activities, adding,



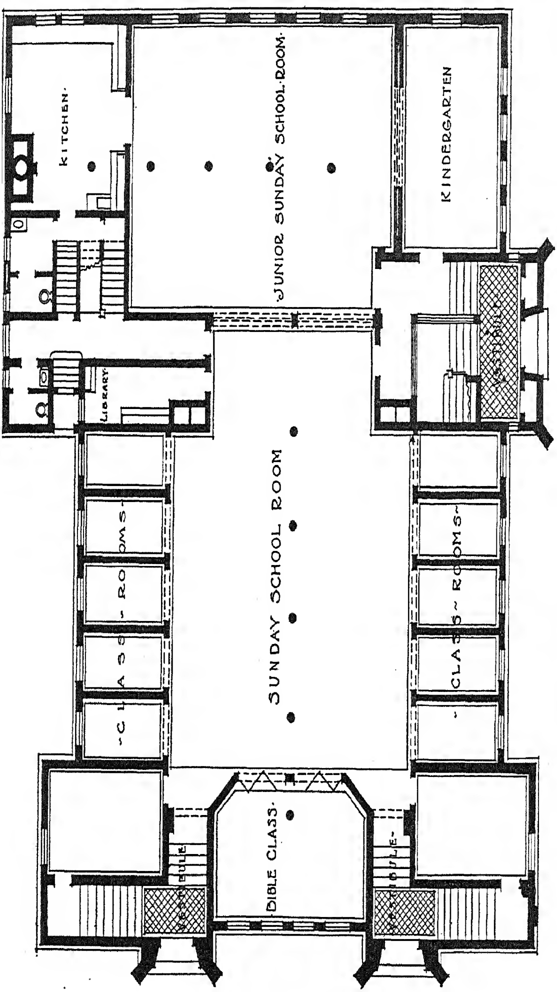
Harry W. Jones, Architect, Minneapolis, Minn.

FIG. 10.—Western (Neb.) Baptist Church

however, considerably to the expense of the building. This church can be built for a modest sum, varying with the material used and the place of construction. An inquiry to the architect will give the information desired.

TEMPLE CHURCH, MINNEAPOLIS

Fig. 11 illustrates another type of medium-cost church which provides an equipment that will relate itself very well to the demands of the graded lessons and social work. Note on the ground floor the departmental rooms for the Primary and Junior departments. What is denominated "Sunday School Room" in the plan serves for general assembly of the Intermediate, Senior, and Adult departments. Ten classrooms aid in providing quiet for the lesson hour. Glass doors in the classrooms admit light to the assembly room. Curtains or flexible doors would divide the main room effectively at the line of posts. The gymnasium, on the floor above the "Junior Room," provides excellent floor space for play and entertainment. The main auditorium provides ideal assembly for worship should the ground floor be needed for additional classrooms or departmental space. In a building of this type, which, by the way, has a pleasing exterior, there is possible every provision for social life for young people. A gymnasium is planned at the rear of the auditorium floor above



Harry W. Jones, Architect, Minneapolis, Minn.

FIG. 11.—Temple Church, Minneapolis, Minn. Ground Floor

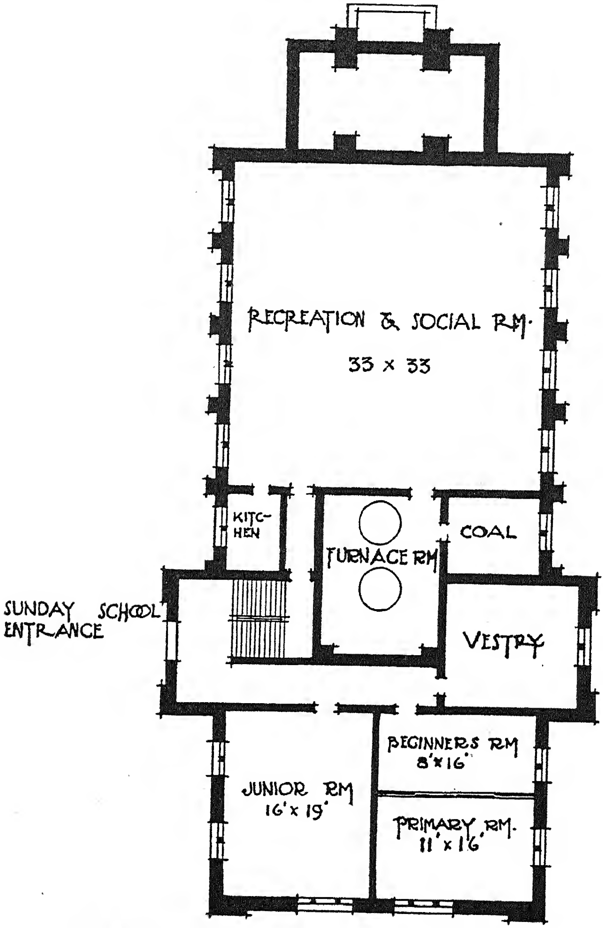
the "Junior Sunday School Room." The graded school will find reasonably adequate accommodations and good opportunity for departmental organization and sessions, five rooms being available for this purpose. This building is considerably more expensive to construct than the preceding, but is within the means of many churches in the country centers.

CANADIAN COMMISSION PLAN

Figs. 12, 13, 14 reproduce a plan offered in the "Report of the Commission on Religious Education" to the Canadian Presbyterian General Assembly. The plans are for a building to care for about one hundred pupils. The longitudinal section (Fig. 12) is suggestive in that it shows that the class or departmental rooms are entirely above ground. The Beginners, Primary, and Junior rooms may be thrown together for worship. This room makes a desirable chapel for other church purposes. The Intermediate and Senior young people have two excellent departmental rooms which, by means of partitions, may be made into four good classrooms. The auditorium is available for worship, which in a small school will include all from the Junior age up. A recreation and social room of generous size and high ceiling provides admirable facilities for play and social life (Fig. 13). There is much that is valuable in

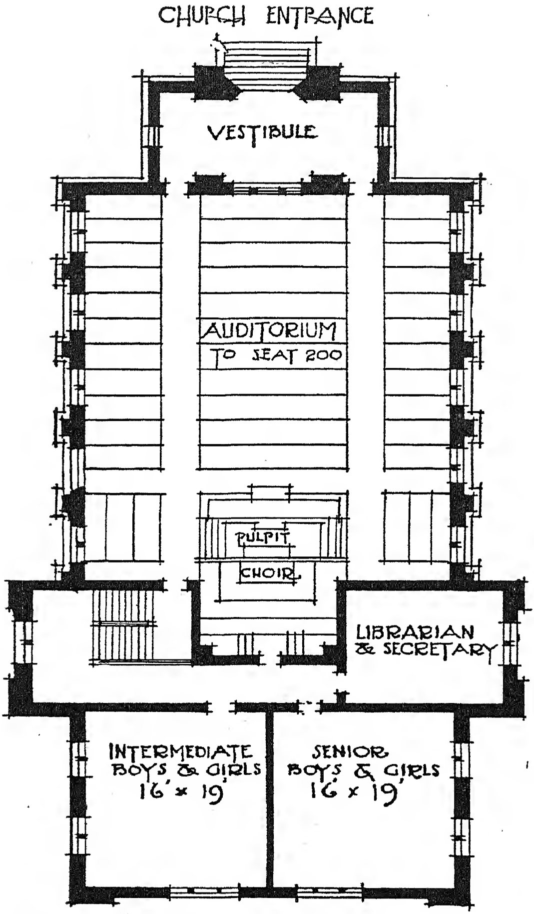
Sharp & Brown, Architects, Toronto, Canada

FIG. 12.—Canadian Commission Plan. Longitudinal Section



Sharp & Brown, Architects, Toronto, Canada

FIG. 13.—Canadian Commission Plan. Ground-Floor



Sharp & Brown, Architects, Toronto, Canada

FIG. 14.—Canadian Commission Plan. Main Floor

this plan. The author ventures the suggestion that with the use of the auditorium for worship, *four* plastered classrooms with separate entrances would add to classroom efficiency.

CHAPTER IX

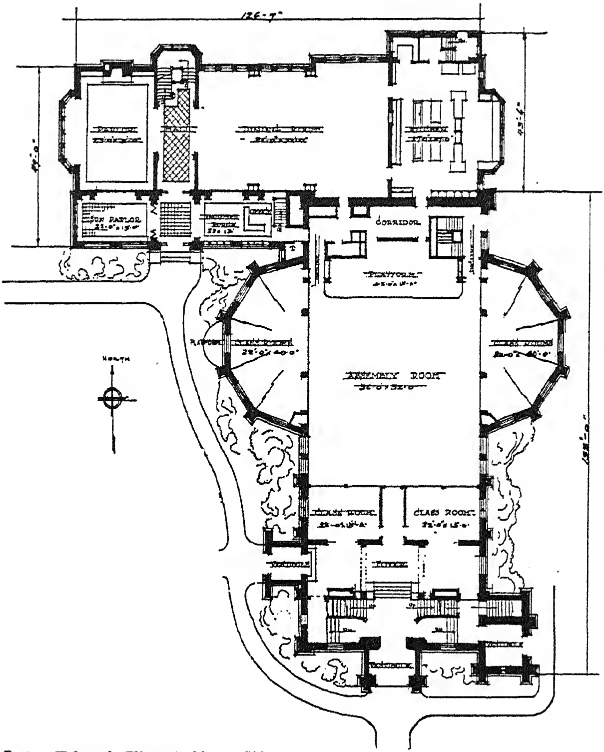
THE SUBURBAN SUNDAY-SCHOOL BUILDING

The "suburban" church problem is recognized as a peculiar type. But when it is approached from the standpoint of community service it does not possess the hopelessness which seems to envelop the church of conventional type. It is possible to develop a community spirit in a suburban situation which cannot be accomplished in a section of a large city. Hence a church which intelligently seeks to serve its community will get a response that will be gratifying. The nearness of the city demands that special attention in many cases be given to the problem of recreation and amusement. The near-by city makes it possible to bring fine talent for aesthetic development to the suburb. Lecturers on civic topics are readily obtainable. Indeed, the leaders in the city often live in the suburbs. Opportunities for social service are abundant in the city so easily reached by convenient transportation. The problem of the suburban church is not insoluble. There are presented below the plans of buildings in two suburbs of Chicago which will prove suggestive and in many cases revolutionary.

OAK PARK CONGREGATIONAL CHURCH

Figs. 15 and 16

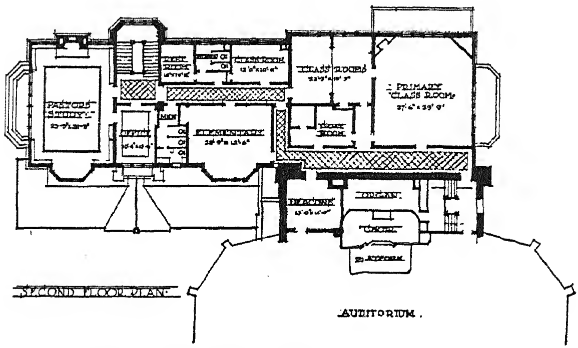
This plan will illustrate two things: the response of this church to the Oak Park suburban situation



Patton, Holmes & Flinn, Architects, Chicago

FIG. 15.—First Congregational Church, Oak Park, Ill. Ground-Floor.

and also the possibilities of remodeling an old building. The ground-floor plan shows the assembly room of the school for the Intermediate, Senior, and Adult classes. Each group of five classrooms is separable from the main assembly room by heavy curtains. This makes possible, for example, a Boys' and a Girls' department. The adult classes adjourn to the dining-room and parlor of the



Patton, Holmes & Flinn, Architects, Chicago

FIG. 16.—First Congregational Church-House, Oak Park, Ill.

"Church-House," as they call their newly constructed addition to their main building. The dining-room may be divided by rolling partitions when necessary into four or six classrooms. The second floor provides a generous-sized pastor's study which is also used as a popular adult classroom. The Primary Department has a large, cheery room, and the Junior or Elementary

Department has a general assembly and adjacent classrooms. Contrary to general usage, the entire *third* floor is planned as a play gymnasium. At present it is unfinished, but is available for this purpose or for additional classrooms whenever needed. There is a large clubroom in the basement, the plan of which is not shown here. A large gymnasium under very competent management is open to the young people of the church in a near-by high-school building, hence that feature is not at present developed in the church. It will be seen that the "Church-House" lends itself admirably to social life, for the young people and for the entire parish. This plan will repay careful study, especially of its provisions for social life.

WINNETKA (ILL.) CONGREGATIONAL CHURCH

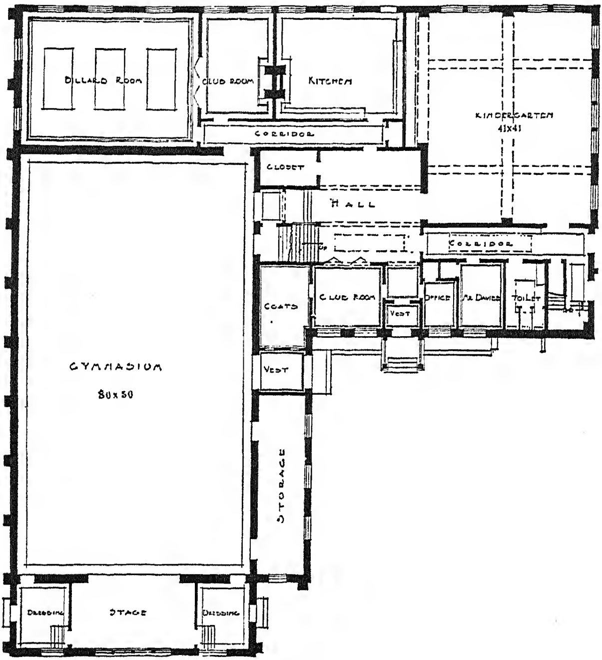
One of the most complete plants for a suburban church in the United States is that of the Winnetka Congregational Church (Figs. 4, 17, and 18). Ideally situated in natural woods, with grounds skilfully handled by the landscape gardener, the approach is all that could be desired. Winnetka is a suburb 17 miles from Chicago and has a population of about four thousand people. A few years ago the church was only a small wooden structure, unattractive in character. Under the skilled direction of a man with a vision, a stone church was constructed which everyone thought

would be adequate for a generation. The splendid graded school within two years overcrowded the new building and for a time was compelled to meet in two separate sessions. The people of Winnetka were pleased with the work of the church and responded generously to a second appeal, giving over a hundred thousand dollars in all for the church and "Community House." (The plans of the Community House only are shown here, Figs. 17 and 18.) Part of the work of the church school is done in the other portion of the church building, which is not shown here.

An excellent room is available on the ground floor (Fig. 17) for the Primary Department. A modern kitchen supplies convenient service to any portion of the first floor. Fully appointed club-rooms are open for men all day and evening. The large gymnasium with high ceiling affords an ideal floor which is busy morning and afternoon all the week, with classes for men, women, boys, girls, and young people. A stage gives opportunity for amateur entertainments. This room is used two or three times a week for motion pictures. The seating capacity of seven hundred is frequently taxed by the people of Winnetka. Only the highest grade of films, locally censored, is ever allowed. So successful is this feature of the work of Community House that no commercial motion-picture theater has located in Winnetka. Community

House is a real center for the people of this little suburban city.

The second-floor plan (Fig. 18) shows ten club-rooms which are occupied week-day afternoons

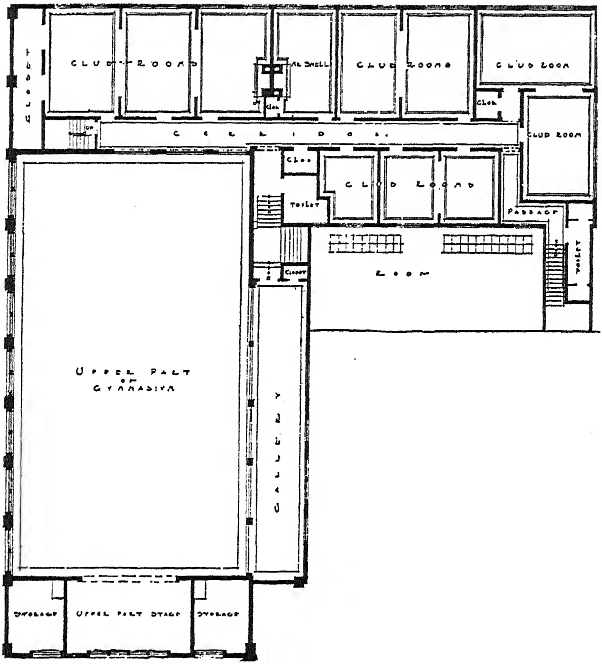


Courtesy of Rev. J. W. F. Davies, Winnetka, Ill.

FIG. 17.—Winnetka (Ill.) Community House. First Floor

and evenings by the boys and young men, and on scheduled occasions by the girls and young women. These rooms are used for class work on Sunday.

The basement plan, not shown here, has ample facilities for private shower baths and locker-room and additional play space that some day may be



Courtesy of Rev. J. W. F. Davies, Winnetka, Ill.

FIG. 18.—Winnetka (Ill.) Community House. Second Floor

used as bowling alleys. In the height of the winter season the weekly attendance at this busy community center exceeds two thousand. Here is a church which believes in serving every need of the

community. Its buildings have become a center of local activity. The two ministers are busy men in the large service that they are rendering. The story of Winnetka Church is an inspiration to anyone who learns of its high degree of efficiency and its extended service in manifold ways to the entire community.

CHAPTER X

THE CITY SUNDAY-SCHOOL BUILDING

More large graded Sunday schools will be found in our cities, and the requirements will call generally for larger and more complicated buildings. In this chapter plans will be shown that seek primarily efficiency for Sunday-school instruction. In the next chapter plans of buildings will be shown which also include facilities for community service. Probably no one building plan will be found suitable to the needs of any given community. These plans reflect the efforts of building committees and architects to meet the peculiar needs of special communities. Their value to the reader lies in their suggestiveness. The ideal building in any community involves consultation with skilled architects. The individual needs will find expression in architectural forms, perhaps allied to some one of these plans, or possibly initiating a partially new type. The ten plans offered in these two chapters represent the very latest efforts of the best architects of this country. The careful study of these plans cannot fail to bring suggestions to the building committee.

SOUTH BEND (INDIANA) METHODIST CHURCH

The plan of the South Bend Methodist Church (Figs. 19 and 20) has some suggestions for those who wish to have an auxiliary auditorium for the Sunday school, rather than to use the church auditorium. Note the effective separation of the Beginners and Primary departments from the main Sunday-school room. Provision is afforded for sex separation in both the Junior and the Intermediate departments. This school is planned for a worship assembly of all from the Junior age upward, Primary and Beginners departments meeting separately. The basement plan, not shown here, includes a large room for dining and entertainment purposes.

SOME KRAMER PLANS

Mr. George W. Kramer, of New York City, has perhaps planned more churches and Sunday-school buildings than any other living architect. His latest work is therefore worthy of careful consideration. Mr. Kramer has always been an enthusiastic supporter of the Akron plan and did much to develop it during the years of the International Uniform Lesson ascendance. He also shows in all of his work the thought of "togetherness" referred to in the paragraph on the Akron plan. By the courtesy of this busy man we are

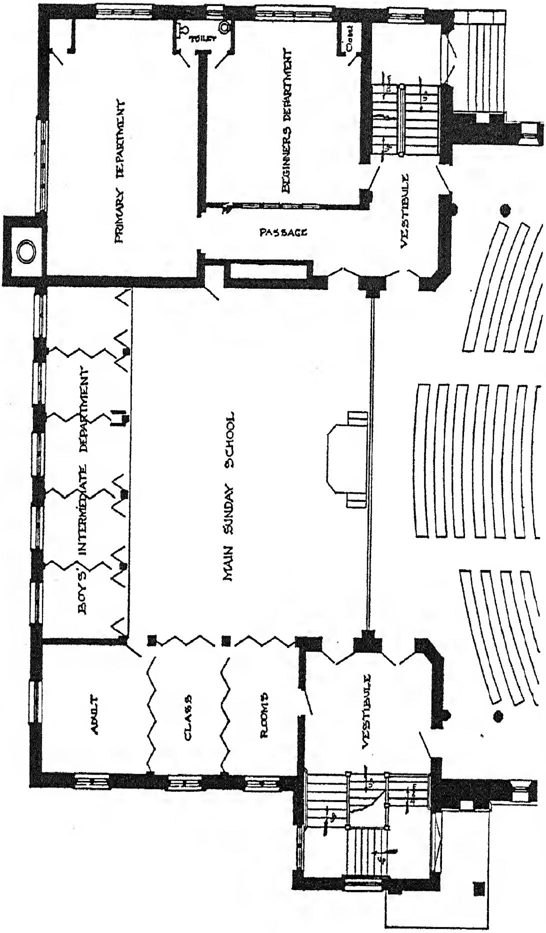
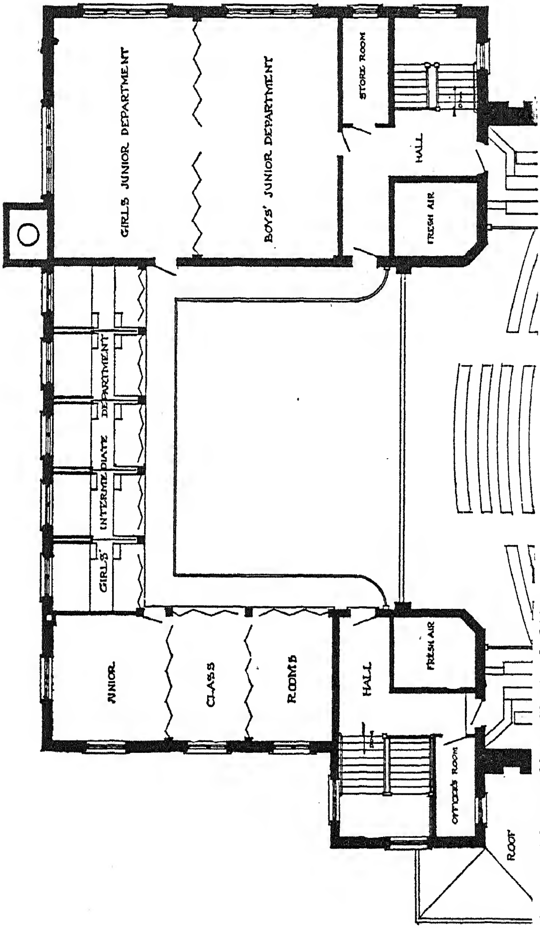


FIG. 19.—South Bend (Ind.) Methodist Church. First Floor

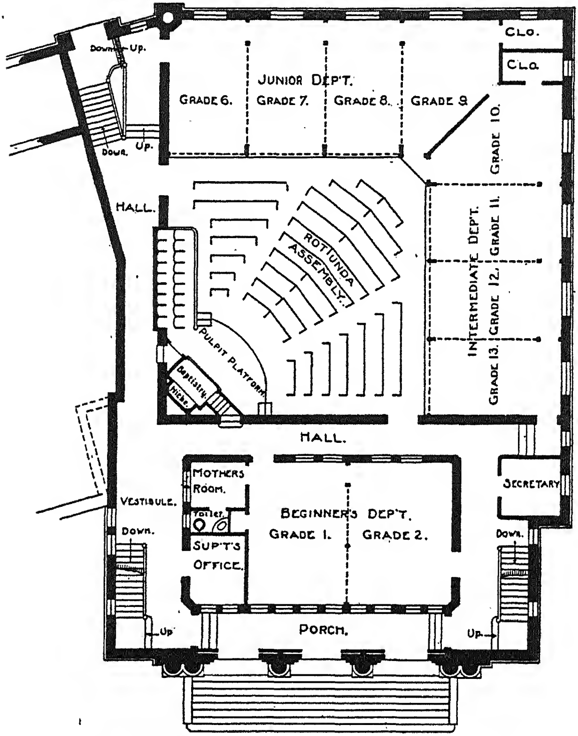
S. R. Badgley, Architect, Cleveland, Ohio



S. R. Badgley, Architect, Cleveland, Ohio

FIG. 20.—South Bend (Ind.) Methodist Church. Second Floor

enabled to examine three of his latest plans, in all of which he had in mind the graded-lesson system.

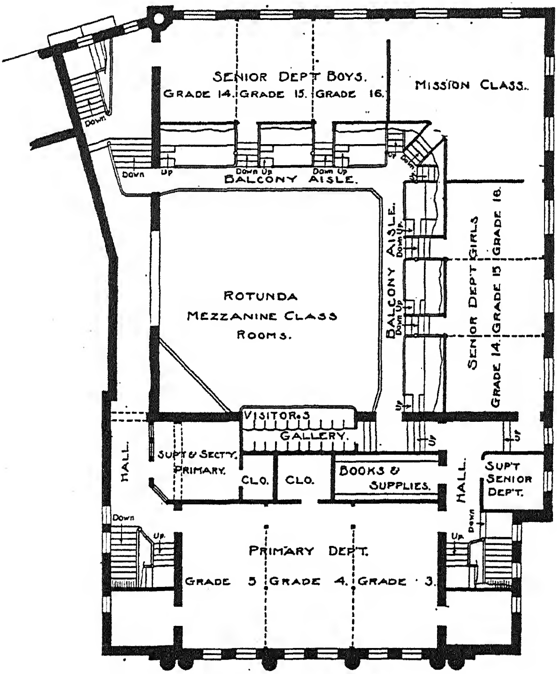


G. W. Kramer, Architect, New York

FIG. 21.—Norfolk (Va.) Christian Church. First Floor

Plan "A" (Figs. 21, 22) is a complete Sunday-school building for the First Christian Church, Norfolk, Virginia, planned for 800 to 1,000 pupils.

The grade notations do not follow the International plan, which does not give a grade number to the Beginners. The departments are well



G. W. Kramer, Architect, New York

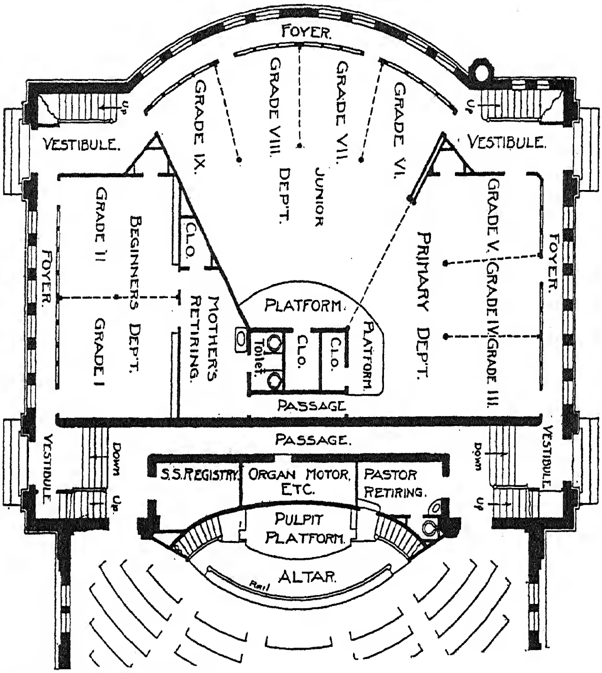
FIG. 22.—Norfolk (Va.) Christian Church. Mezzanine Floor

segregated, and provision is made for general assembly in what is now the auditorium of the church. Note the sex segregation in the Senior

Department classrooms which are installed under a deep balcony. Additional classrooms are provided adjacent to the Primary Department which is on the second floor. The third-floor plan, not printed herewith, shows the deep gallery and two large rooms for classes or social life.

Plan "B" (Figs. 23, 24, 25) was made by Mr. Kramer for the Methodist Episcopal Church South, of Conway, Arkansas, and is regarded by him as "one of the best types of arrangement for departmental schools." The plan combines all in two groups for worship, segregates the Beginners Department, arranges for assembly of Junior and Primary departments if desired, and provides for separation of both grades and sexes in the Intermediate and Senior departments. The diagonal lines between departments indicate sound-proof movable doors. The whole school can be thrown together into two sections in a moment by raising these doors. The Akron plan is used to provide classrooms in some of the departments. In this plan entrance to all the classrooms is from an outer passage which is also an insulation against outer noises and heat but at the same time provides ample light and ventilation. Note in all of Mr. Kramer's plans the adequate provision he makes for easy passage from department to department, and for convenient exits.

Plan "C" (Figs. 26, 27, 28) is the Jefferson Street Church of Christ, Buffalo, New York. The basement of this church provides large rooms

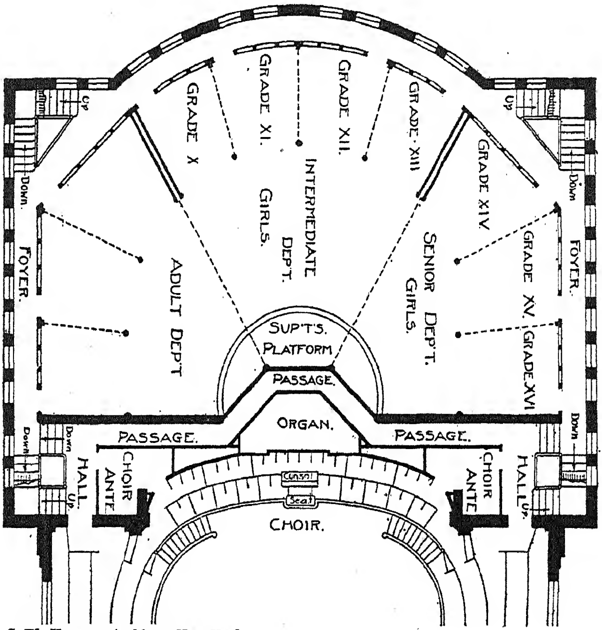


G. W. Kramer, Architect, New York

FIG. 23.—Plan "B." First Floor

for the Beginners and Primary departments *en suite*, but separable when desired by rolling partitions. A gymnasium with separate locker-rooms

for boys, men, and girls occupies a large portion of the space in this high, well-lighted basement. The parlors are available for classroom purposes. The main floor provides separate classrooms and

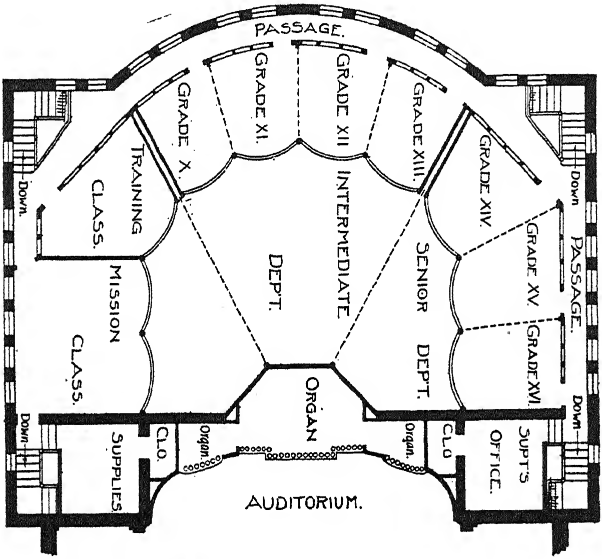


G. W. Kramer, Architect, New York

FIG. 24.—Plan "B." Second Floor

department rooms for Intermediate and Senior boys and girls. The Junior boys and girls occupy the floor space in the center, not an ideal arrangement for this important department. The second

floor provides excellent classrooms for organized classes. This plan will be attractive to those who wish to gather the entire school above the Primary Department into one worship session.

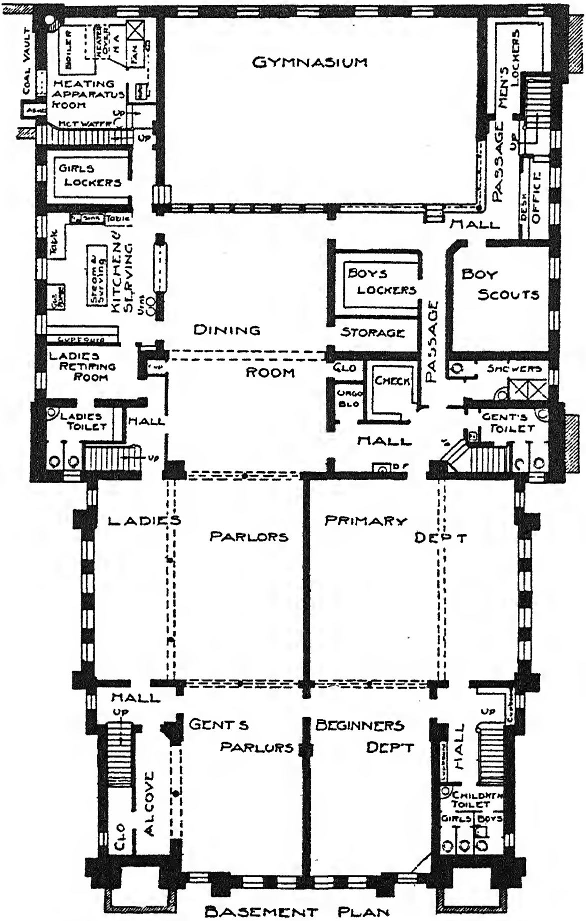


G. W. Kramer, Architect, New York

FIG. 25.—Plan "B." Third Floor

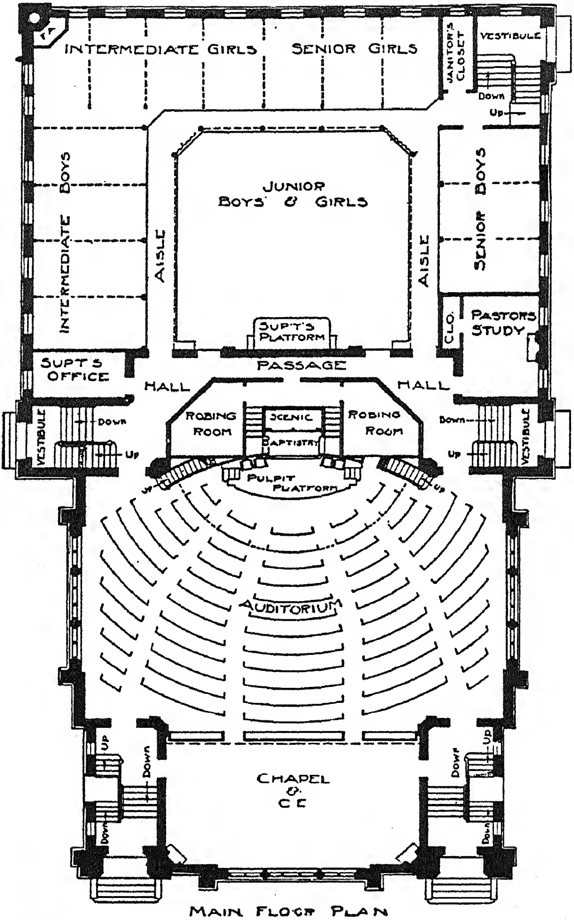
SAN DIEGO (CALIFORNIA) BAPTIST CHURCH

This new church (Figs. 29, 30, 31), recently completed at a cost of about one hundred thousand dollars, presents some helpful suggestions. Fig. 29 shows the basement floor, well lighted. The



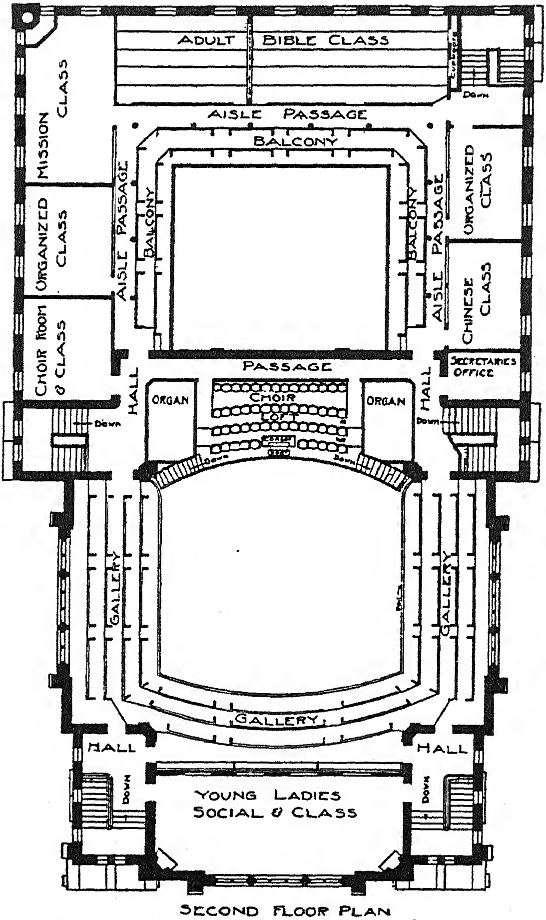
G. W. Kramer, Architect, New York

FIG. 26.—Jefferson Street (Buffalo, N.Y.) Church of Christ



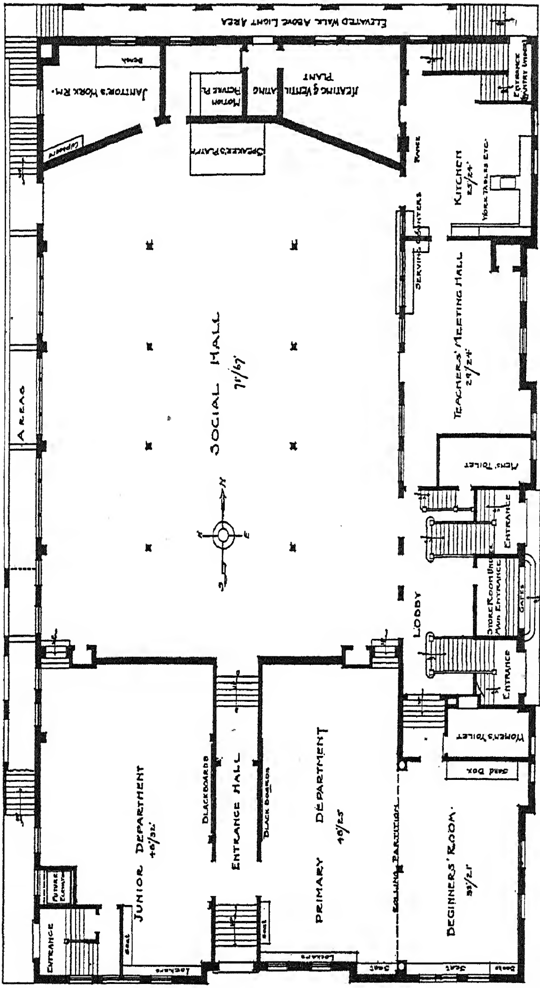
G. W. Kramer, Architect, New York

FIG. 27.—Jefferson Street (Buffalo, N.Y.) Church of Christ



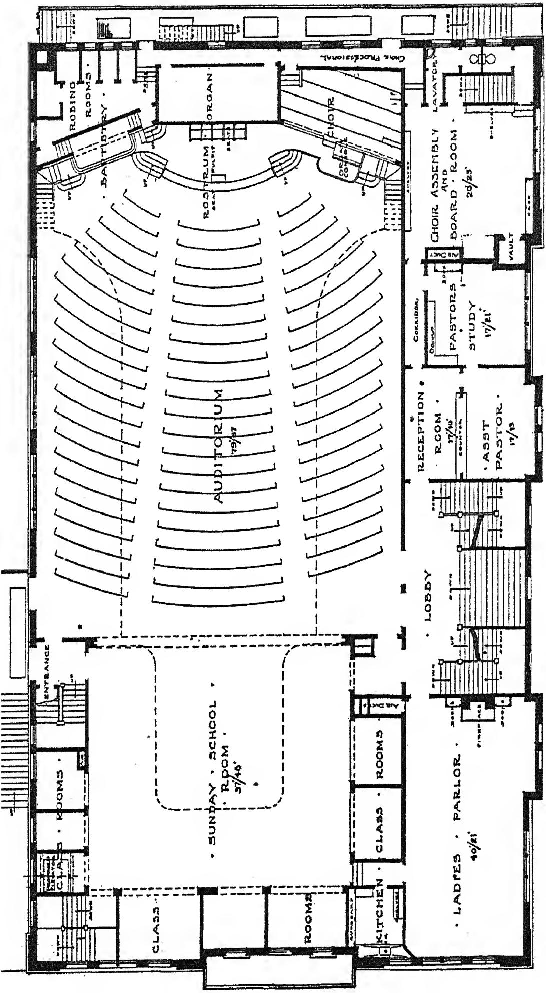
G. W. Kramer, Architect, New York City

FIG. 28.—Jefferson Street (Buffalo, N.Y.) Church of Christ



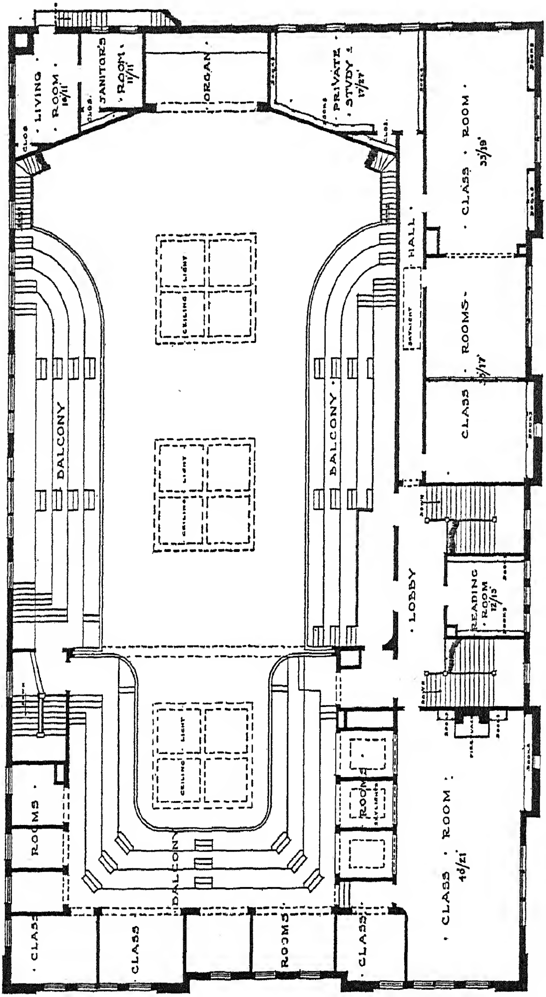
Norman F. Marsh, Architect, Los Angeles, Cal.

FIG. 29.—San Diego (Cal.) Baptist Church. Basement Floor



Norman F. Marsh, Architect, Los Angeles, Cal.

FIG. 30.—San Diego (Cal.) Baptist Church. Main Floor



Norman F. Marsh, Architect, Los Angeles, Cal.

FIG. 31.—San Diego (Cal.) Baptist Church. Second Floor

Beginners and Primary departments have large separate rooms which may be united by opening rolling partitions between. A near-by toilet is convenient to this department. Across a wide hall, that acts as an effective barrier to noise, is a roomy Junior Department. This is broken up into classrooms by large, heavy, movable, partition screens, planned with a broad footing especially for this purpose. The social hall and dining-room is planned for all social occasions. A fireproof motion-picture equipment is built into this room. A special feature, worthy of praise, is the "teachers' meeting-hall," adjacent to the kitchen. The supper hour is proving to be an excellent time for the teachers' meeting. On the main floor is the auditorium and the Intermediate and Senior departments. The latter is provided with classrooms, both on the main floor and in the balcony. In addition there are several large rooms for organized classes and social life. The building is one of the best of the type that requires a combination auditorium for church and Sunday school.

The next chapter will present other plans for city churches in which special provision is made for community service.

CHAPTER XI

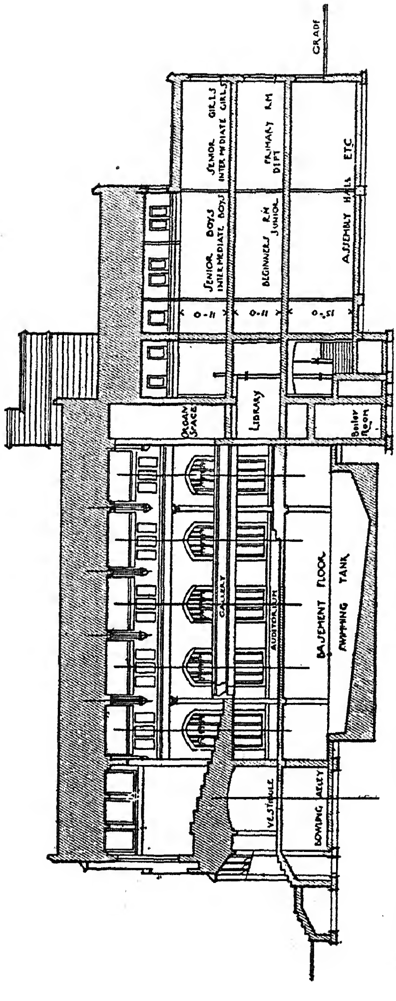
THE CITY SUNDAY-SCHOOL BUILDING, WITH SPECIAL REFERENCE TO COMMUNITY SERVICE

The object of the Sunday school is to aid the pupils to achieve a Christlike character which will relate itself to the well-being of the race. The modern city has multiplied the influences against the highest type of character to such a degree that the church must broaden her efforts to save boys and girls to the higher life. It is not sufficient to plant the Christ ideal at the psychological moment; it is necessary also to supply as far as possible the wholesome environment in which the Christ ideal may develop to its full normal maturity in a strong life characterized by self-control and achievement of the highest and the best the race knows. It is not enough to have an hour's session Sunday mornings, no matter how efficiently the instruction is given. The challenge that comes to our city Sunday schools today is a far larger one. Wherever there is a lack in the environment of our youth it is the opportunity and duty of the agencies of religious education to see that the need is met. The leisure hours of boys and girls are most prolific for good or evil. Voluntary

interests have largest play in these hours. The church which seeks to direct the leisure time of her youth is in line with the best thought for character development. The more the youth's interests are centered in the church building, the more certainly may the youth be won for Christ and for life's highest ideals. The plans which follow emphasize in a special degree the conception of a church building put to the largest use in the great task of developing a kingdom of righteousness among our young people.

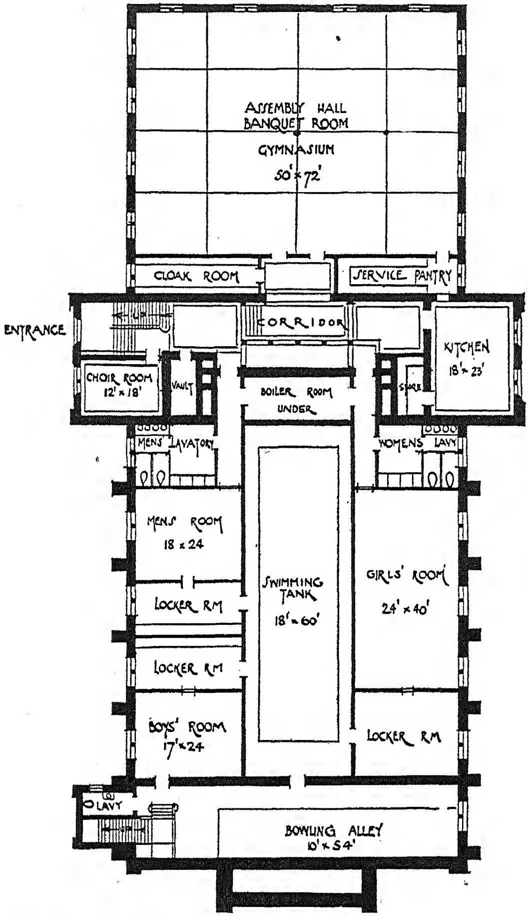
CANADIAN COMMISSION PLAN

The plans for this building (Figs. 32, 33, 34, 35) are for a school of a membership of two to five hundred pupils, exclusive of adults. The auditorium of the church may be used for the worship of Intermediate, Senior, and Adult departments. Adult classes will remain in the auditorium for the study hour while the Intermediate and Senior classes retire to second-floor classrooms adjacent to the church gallery. In both of these departments separate rooms are provided for girls and boys. Within each of these rooms provision is made for separation into class groups by means of rolling partitions. The inner classrooms have overhead light and ventilation. Young men and young women have excellent rooms provided for their class sessions and other activities. The



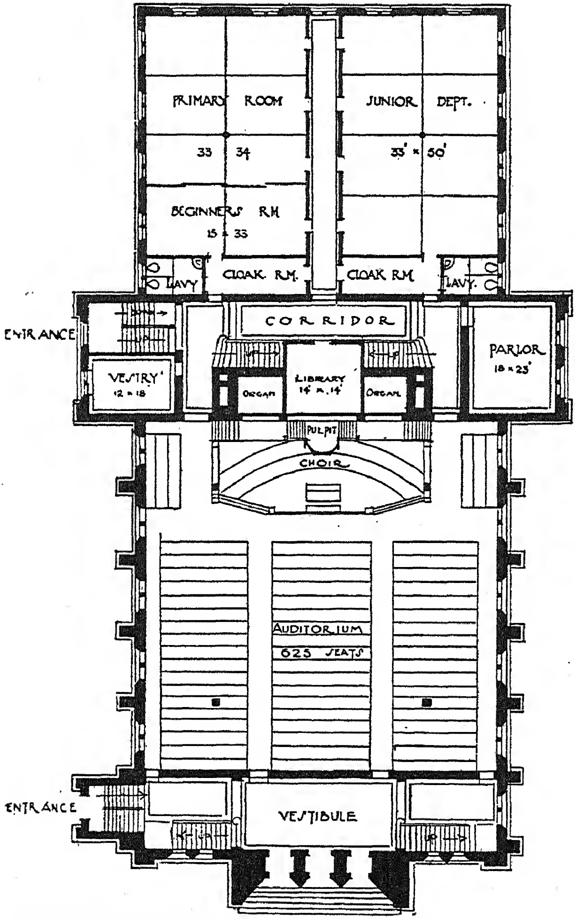
Sharp & Brown, Architects, Toronto, Canada

FIG. 32.—Canadian Commission Plan. Longitudinal Section



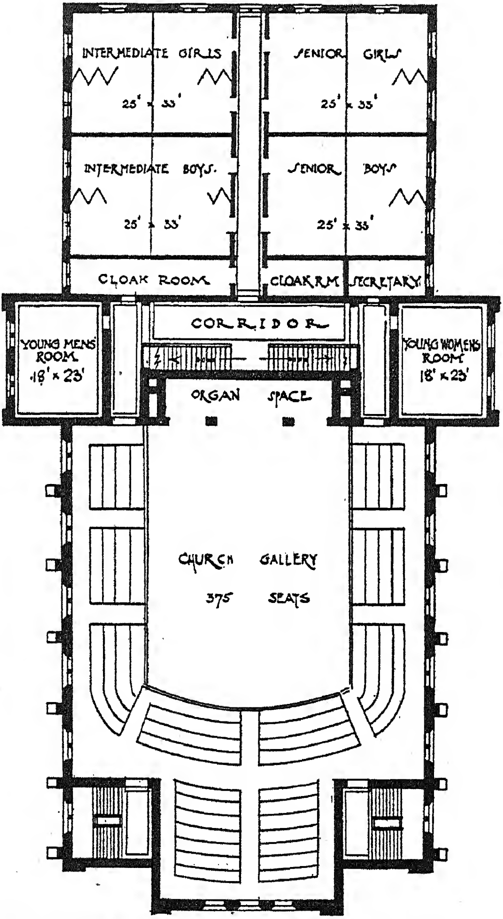
Sharp & Brown, Architects, Toronto, Canada

FIG. 33.—Canadian Commission Plan. Basement



Sharp & Brown, Architects, Toronto, Canada

FIG. 34.—Canadian Commission Plan. Main Floor



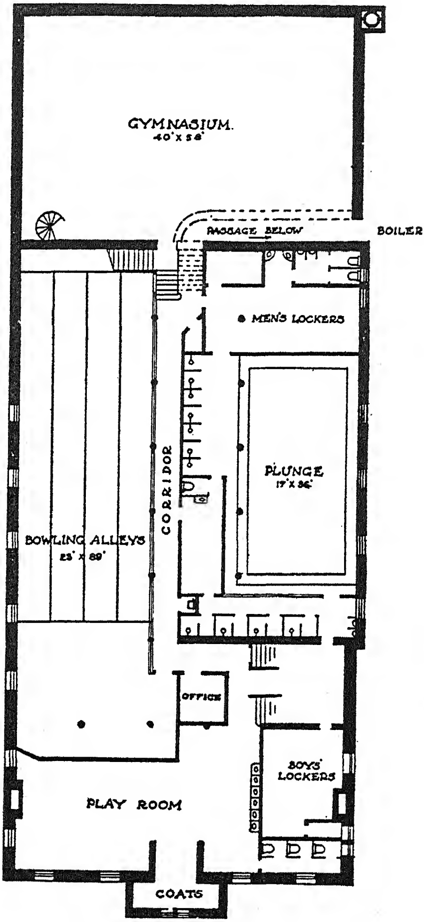
Sharp & Brown, Architects, Toronto, Canada

FIG. 35.—Canadian Commission Plan. Second Floor

ground-floor plan (Fig. 34) shows ample provision for Beginners and Primary departments, which may be thrown together on occasion. Separate class-rooms also are planned by means of curtains or rolling partitions. Across a corridor, which effectively breaks disturbance by sound, is a Junior Department. Separate assembly is possible, and usually is desirable. There is also easy access to the auditorium of the church. Each grade can be segregated for instruction by rolling partitions. Note the excellent provision for wraps on both floors, also the lavatories for both Primary and Junior children. The basement-floor plan (Fig. 33) shows a good assembly hall and gymnasium, 50×72 feet in size, with a 15-foot ceiling. Separate rooms are provided for club life for boys, men, and girls. Separate locker-rooms adjoin each of these club-rooms. A bowling alley and swimming tank complete the equipment of this well-planned floor for community service.

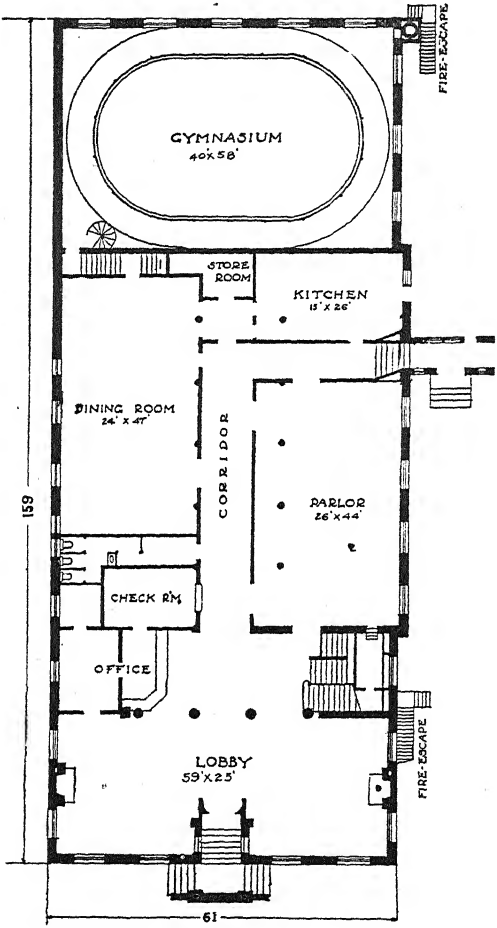
ROCHESTER (NEW YORK) BRICK CHURCH INSTITUTE

Figs. 36, 37, and 38 reproduce the floor plans of a building adapted to a specialized form of community service which is greatly to be commended. The plans, as a whole, are those of a high-grade city Y.M.C.A. building. The basement (Fig. 36) contains a standard gymnasium, four bowling alleys, showers and plunge with lockers for men,



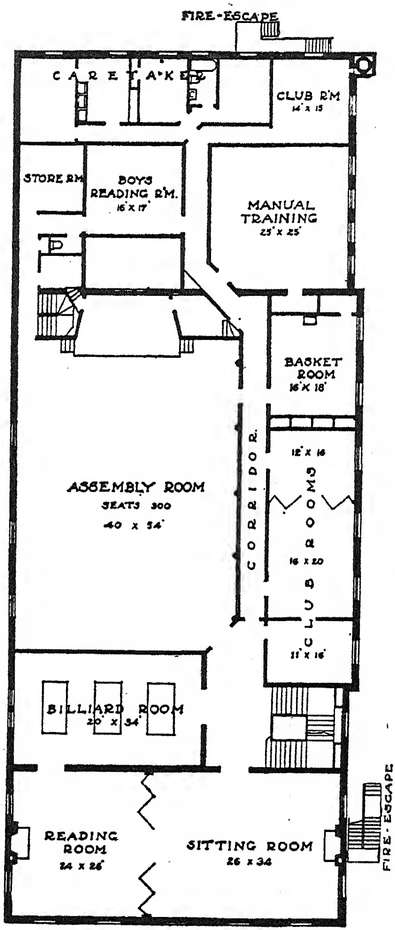
Courtesy of Rev. Herbert W. Gates, Rochester, N.Y.

FIG. 36.—Brick Church Institute, Rochester, N.Y. Basement



Courtesy of Rev. Herbert W. Gates

FIG. 37.—Brick Church Institute, Rochester, N.Y. First Floor



Courtesy of Rev. Herbert W. Gates

FIG. 38.—Brick Church Institute, Rochester, N.Y. Second Floor

and separate playroom with lockers for boys. The first floor (Fig. 37) has a generous lobby, parlor, dining-room, and kitchen. These latter facilities are quite effective in work for young men. The second floor provides for the varied interests of work with and for young men and boys. The plan (Fig. 38) will explain itself. The entire third floor, not shown here, is a dormitory providing numerous bedrooms for young men. The adaptability of this building to the religious education of boys and young men will be apparent without detailed description. Assembly rooms and classrooms are available for every need. Most of the Sunday school of this church is cared for in the church building adjacent. This plan is shown to illustrate a form of specialized community service of the highest value. Let any church ask itself whether the young men of the city need a *home* and a club under church influences.

THE CLEVELAND PLAN

A plan has been prepared by a firm of Cleveland architects which has many excellent features. We regret that suitable arrangements could not be made to exhibit it in this book. It will be described, however, and anyone with a good imagination can reproduce the plan with pencil and paper. The building is rectangular and provides one auditorium for Sunday-school and church worship on the

ground floor. Provision is made for departments and classrooms at the rear of the pulpit platform. The Junior Department is on the ground floor and is divisible into eight classrooms by accordion doors. Each classroom has a separate entrance to the wide corridor. The auditorium-balcony plan provides for two departmental rooms for Senior and Adult classes, each of which may be divided into four classrooms with separate hall entrances. Two larger classrooms adjacent to the balcony of the church auditorium are provided. The basement has both a floor and a balcony plan. The floor plan shows a gymnasium and entertainment hall, with Beginners and Primary departments opening from either side by accordion doors. At one end of the gymnasium is a platform and at the other are showers and lockers situated under a roomy balcony. A full-sized bowling alley, with seats for spectators, occupies the space under the tier of classrooms described above. The basement-balcony plan provides for an Intermediate Department above the bowling alley, separable into eight classrooms, each of which opens into a long corridor.

The method of providing departmental rooms separable into classrooms by accordion doors has been in use for several years. The Long Beach (California) Methodist Episcopal Church, of which Norman F. Marsh, of Los Angeles, is the architect, plans its Junior Department in precisely this

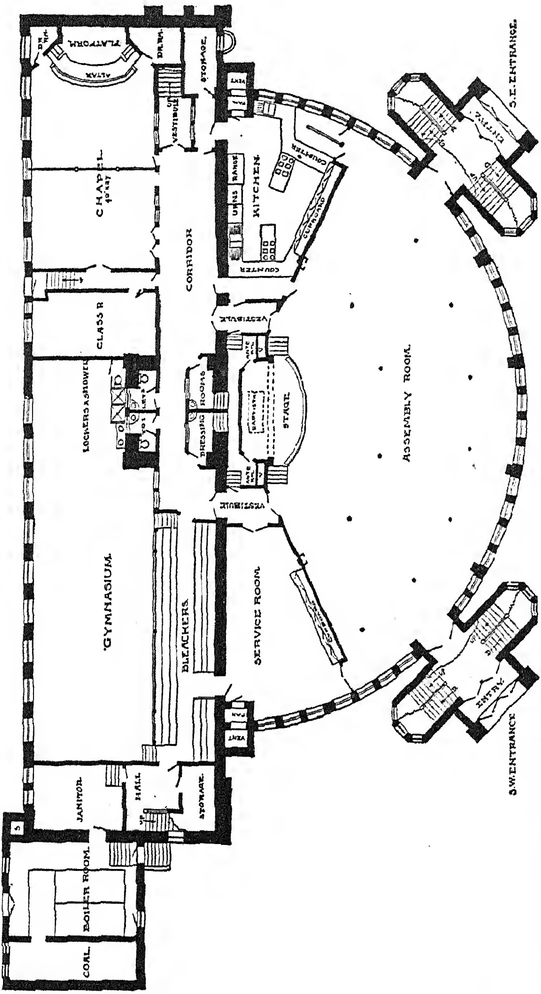
manner. The use of accordion doors for the separation of classes cannot, however, be regarded as *ideal*, as our earlier discussion has pointed out. The ideal plan should show a larger number of plastered classrooms. This could be accomplished, with slight change in this plan, by making half of all classroom partitions permanent. Departmental assembly is secondary to classroom efficiency. Many would object to the placing of the Beginners and Primary departments in the basement, but this is a detail of arrangement easily changed by shifting adult or young people's classes to the less desirable rooms. The type of plan described above in which the church auditorium is used for Sunday-school worship is sure to be used largely in coming years. This will be illustrated in the next section by the plan which is, to the present time, nearest the ideal.

THE CEDAR RAPIDS PLAN

Several years ago St. Paul's Methodist Episcopal Church, Cedar Rapids, Iowa, facing the necessity of a new building, made a special investigation of church buildings available at that time, and conducted a wide correspondence with Sunday-school experts and architects. Many architects were invited to participate in the competition. Much detailed information obtained from many sources was sent to each contestant.

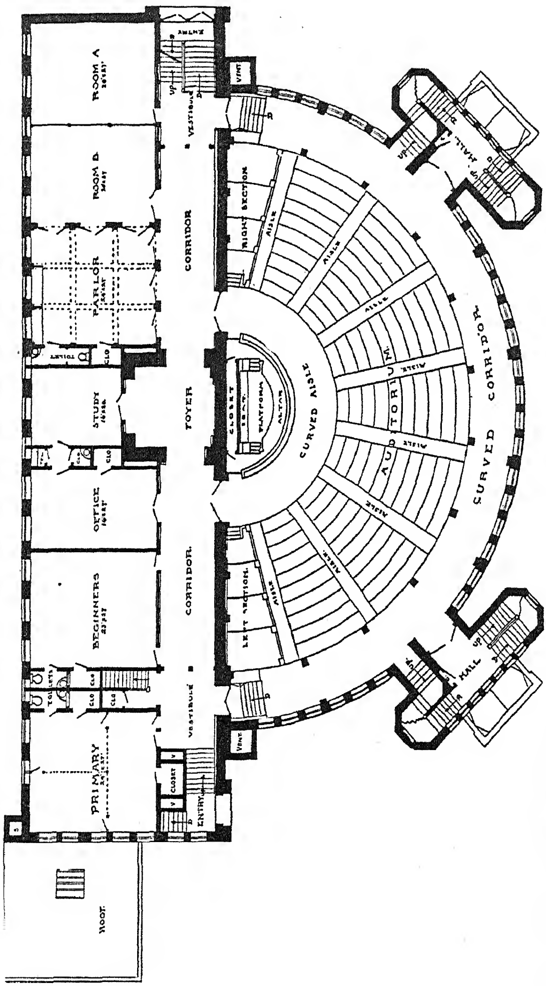
The competition was won by Mr. Louis H. Sullivan, of Chicago, one of America's leading architects. The building was not constructed by Mr. Sullivan, but the floor plans, substantially as prepared by him, were used by the church, the exterior being changed.

This plan (Figs. 39, 40, 41) is presented last because it is regarded by many, including the author, as probably the most significant contribution to the architecture of the modern Sunday school made to the present time. It is worthy of the most careful study by any prospective church builders. It was born of a desire to make more adequate provision for the ages when youths most rapidly leave the church. With the exception of the Beginners and the Primary and perhaps the Junior departments, worship for all is planned in the church auditorium. Separate plastered classrooms are provided for every class in the school except in the Junior Department, where removable partitions are used. The classrooms correspond to the ideal outlined in chap. vi. Unusually wide corridors provide for social life, and for the delay which may occur when the Sunday-school and church services approach one another. A beautiful chapel provides for devotional meetings and for departmental assembly. An assembly room with stage, in the basement, is available for entertainments. A gymnasium, with gallery for



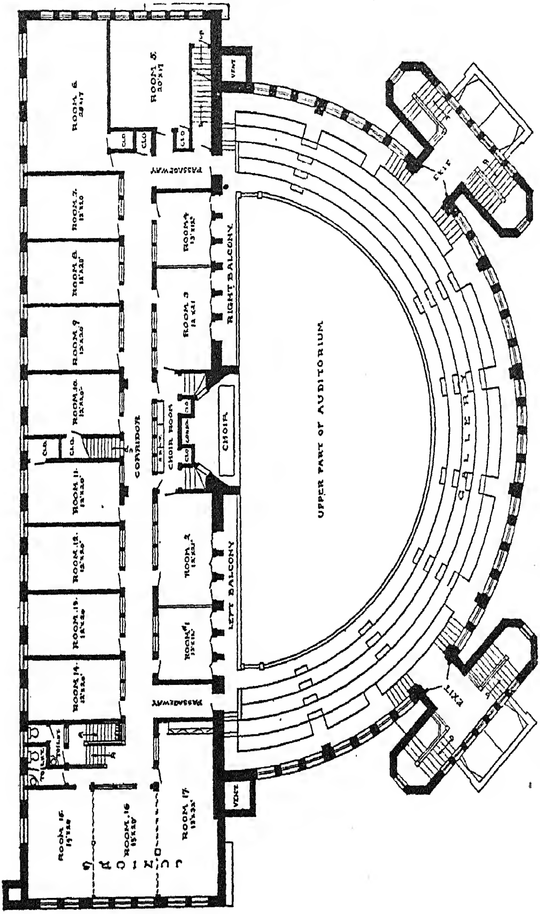
W. C. Jones, Constructing Architect, Chicago

FIG. 39.—St. Paul's Methodist Church, Cedar Rapids, Iowa. Basement



W. C. Jones, Constructing Architect, Chicago

FIG. 40.—St. Paul's Methodist Church, Cedar Rapids, Iowa. Main Floor



W. C. Jones, Constructing Architect, Chicago

FIG. 41.—St. Paul's Methodist Church, Cedar Rapids, Iowa. Second Floor

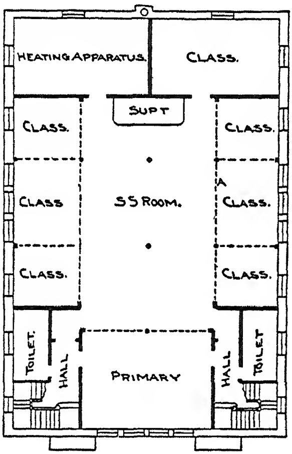
seventy-five spectators, provides for the play life of the youth. This plan promises to influence future Sunday-school construction in a marked degree. Its enthusiastic supporters call it the "Cedar Rapids plan," and predict that it will have the vogue in the next quarter-century which the Akron plan has enjoyed in the last twenty-five years.

CHAPTER XII

REMODELING OLD CHURCH BUILDINGS

It frequently occurs in the history of a growing church that the Sunday school crowds its quarters. In some cases, especially in country and village churches, the building is little more than one large bare room. The building may be substantial in construction or the congregation may be unable to rebuild to satisfy the modern demands. What can be done? Obviously each problem of this type is individual, not permitting of a general answer. However, certain suggestions can be made which will help in making over the old structure into something more modern. The study of the best plans, such as have appeared in this book, will indicate the type of building which is desirable. The competent church architect will be able to accomplish much more than perhaps seems possible. Does the old church have a high and dry basement? This may provide a quiet room for the Beginners and Primary departments by means of plastered partition walls, while six to a dozen classes may have good rooms by means of the temporary curtains on wires or brass rods, or the more permanent rolling partition (see Fig. 42). Where sufficient money is available for an addition

it is usually advisable to use the funds for the Sunday-school quarters, for the modern demands are relatively so complicated that it would be better to build a new Sunday-school building than to attempt to alter an old churchbuilding into Sunday-school quarters. There is often a lack of light



G. W. Kramer, Architect, New York

FIG. 42

in the old church building which will require window alterations. Usually more can be accomplished by building the Sunday-school portion new. When this is the case, ideas and suggestions will come from the late plans offered herewith. For instance, note how effectively a modern building

can be realized by adopting the Cedar Rapids plan, using the old auditorium for worship and adding classroom facilities.

The same principle can be adopted in a modified form in the smallest church. The writer recently saw in California a church alteration costing about a thousand dollars which had transformed the Sunday school from a one-room organization to a three-department school with three additional classrooms. This was accomplished by the simple expedient of an addition at the rear of the church, which provided separate rooms for the Primary and Junior departments, and the classrooms mentioned. The membership of the school was about a hundred, and graded work was being used in part. In a large city school, where the problem of classrooms had become acute, a neighboring flat building was rented; unexpectedly efficient quarters for about twenty-five separate classrooms were thus added. A covered sidewalk to the church made the building a constituent part of the Sunday-school plant.

It need never be considered impossible to improve greatly an old building. Even in the case of the single-room country church with no basement, it is possible by means of curtains to add greatly to the efficiency of the school; while one rolling partition will give a Primary Department which will enable the teachers to do infinitely

better work. The cost of curtaining off a half-dozen classes and putting a rolling partition or folding doors across a small building for the Primary Department need not exceed $100. This method will leave the building intact for other purposes. A way can be found when the need is realized.

Two proposed alterations of a more ambitious character may be described in which efficient use was made of the present buildings and at the same time modern equipment was provided for the Sunday school. In each case a large saving over new construction was effected. In the first plan the old building consisted of an auditorium with a Sunday-school room in the rear. The old Sunday-school room was used largely for the Junior Department, and a portion of it for Senior classes. The new construction was two stories—first floor and basement. On the first floor were provided parlors, dining-room, and kitchen, which were also used as classrooms. The Primary and Kindergarten departments were provided with adequate rooms, while the Intermediate Department had excellent quarters with six good classrooms. On the basement floor was planned a 40×40 foot gymnasium, swimming pool and lockers, bowling alleys and clubroom. What a transformation from a two-room, old-style church!

In the second plan a substantial one-room church with a basement Sunday-school room was

transformed by the erection of a two-story and basement addition. The basement plan provided for a dining-room and entertainment room, check- and locker-rooms, and a 30×50-foot gymnasium in the new part. The first floor provided for Kindergarten and Primary departments and parlor, all three capable of being thrown together for social purposes. The Junior Department had an excellent assembly-room and four classrooms. The second floor provided nine classrooms and club-rooms. A delighted people will move into their *new* church, for such it will be with these admirable additions to its equipment. Consultation with a competent church architect will often reveal possibilities of improvement not realized by the layman.

CHAPTER XIII

SUGGESTIONS FOR THE BUILDING COMMITTEE

To the building committee of a church is assigned a most difficult and responsible task. The average church does not construct more than one building in a generation. It is of the utmost importance that mistakes should be avoided and that the new building should be responsive to future needs.

The building committee should enter upon its task first as a *commission* to study the needs, present and future, of the church. This task calls for broadmindedness. The church of the future will be called upon to serve in other ways than the church of the past. The committee should certainly include in its membership some of the younger generation. The question of the amount of the contribution should not be the basis for choice of building-committee members.

The Sunday school is destined to be of increasing importance in the work of the church. The activity of the church in religious education has resulted in the past in three-fourths of the increase in church membership. There is no reason to believe that the coming generation will show a smaller proportion for this division of the church's

work. But in addition to this appeal to the desire for self-perpetuation is the challenge to meet more adequately the community needs for religious and moral education. Our churches through their schools must do more of this necessary work and must do it better in the future. The all-pervasive principle of efficiency demands this larger result. The wise building committee, whether the church is large or small, will give special thought to generous and adequate provision for the Sunday school.

Community service is a new note in our church life which will receive large attention in the coming generation. An earlier chapter (chap. vii) has enlarged upon this theme. The building committee will consider with the utmost seriousness what facilities shall be provided for this field of church activity. Community service keeps the church plant busy more hours every week and relates the church more vitally to the physical, social, educational, and recreational needs of the community.

One of the first and most important steps for the committee to consider is the choice of the architect. The wisdom of the committee in this matter will determine largely the success or failure of the building to be constructed. Shall he be a local man? Shall he be selected by competition? Or shall he be a church specialist, widely informed,

exceptionally competent to help in the problem of the committee? The obvious answer is, The architect should be a church specialist, a man with *wide* experience in building churches, alert to the modern needs. He should not be selected by competition; such a method is unsatisfactory and the best men, except in unusual cases, refuse to enter competitions. The church specialist *may* be a local architect, but there are not many competent men of this type and they are not to be found in every community.

The architect having been selected, he should be treated with the same confidence which the medical specialist or expert lawyer receives. The committee should tell him what it considers its present and future needs, but should be open to his suggestions. He is in close touch with the best that is being done in the country. The committee may well feel free to express to him any preferences in ideals and plans which have come to it in its study.

It is refreshing to learn occasionally of a church building committee that seeks honestly and sympathetically to learn the real needs of the Sunday school, and which recognizes that the future church will be recruited largely from that organization. Building a modern Sunday-school and church building is one of the most complicated tasks the architect is called upon to undertake, for

the transition situation in the Sunday-school world makes difficult the satisfaction of every need, present and future. Many a building constructed within the last four or five years fails to show a suggestion of attempted response to the needs of the modern graded Sunday school. And in many cases the failure lies at the feet of the church building committee, which did not include in its membership representatives of the Sunday school. Despite the radical demands of the new Sunday-school building, every department of church activity can have facilities for its work as good as, or better than, in the older type of building.

Perhaps one of the chief matters of adjustment will be the favorite plan of using the Sunday-school quarters for an extension of the normal audience room. This plan will not be popular in the future. Careful study of many cases has shown that the added seating capacity is rarely used, hence there is no valid reason in those cases that Sunday-school facilities should be sacrificed to the desire for an enlarged auditorium two or three times a year. It is questionable whether a thoroughly effective modern Sunday-school building can be constructed, at the same time making the space available for added seating capacity for the church auditorium.

A new church costing $100,000 recently examined by the author is not less than 40 per cent inefficient for Sunday-school purposes because

the minister insisted on using the Sunday-school room to make an additional capacity of four hundred for his audience room. Another church, costing over $150,000, advertised as the most modern church in its section, has extended the steep church auditorium gallery about the Sunday-school room and used pews of the same style in the Sunday school as in the church proper. And the Sunday school burdened itself through a period of years to pay thousands of dollars toward this building! Whenever a church gains the vision of efficiency in religious education and provision for the leisure hours of its youth, there will be no difficulty in constructing a Sunday-school building which will respond to the new ideal.

MEETING THE NEEDS OF OTHER CHURCH ORGANIZATIONS

An examination of the plans and descriptions preceding will show that while the Sunday school often has been apparently the primary thought, yet other church needs have been amply cared for. What better use can be made of the church parlor, for example, than to make of it a cheerful Primary and Beginners' room for the children? The gymnasium and entertainment room will be found available for the occasional dinners of other church organizations. The secondary auditorium or one of the departmental rooms will serve

admirably as a chapel. Classrooms respond to the needs of committee meetings. Every club will have ample quarters in the classrooms. The possible clash between Sunday school and morning worship, when the same auditorium is used, can be avoided by Sunday-school worship being held at the beginning of the Sunday-school hour with dismissal from the classes and no return to the church auditorium. Dismissal of the Sunday-school groups directly from their classes without closing exercises is a proved success and gives to the individual teacher the opportunity for the last impression. Five to seven minutes of lost time for reassembly is also saved to the lesson.

GENERAL SUGGESTIONS

The basement of the church building is not the ideal place for the Sunday school. If its use can possibly be avoided, efforts should be made to do so. A ground floor with full-size windows is very desirable. This will enable little children to enter their departments with few or no steps. If a basement must be utilized, let the men's classrooms be put there. What true father would consign his children to the basement while he and the mother chose the sunny, cheerful rooms for themselves! Absurd as it may seem, one of the otherwise good plans for a new building, which has recently been constructed, shows precisely that

situation—down the dark stairs for the little children, and a very large east and south room on the ground floor for the "men's class"!

It is better to use leaded, clear glass in Sunday-school classrooms than deep-colored glass. Keep the rooms bright and cheerful. The competent architect will provide good ventilation and light for every room in which people are asked to remain for any length of time. There are technical standards in these respects which should be observed.

Care should be taken that halls are ample and well lighted, that stairs should have an easy tread, and should in no case be of a winding character with narrower footboards at one side than the other. Handrails are desirable in some cases, with a second rail for small children. An adequate sanitary drinking-water supply should be provided. Convenient cloakrooms adjacent to each department are desirable in which umbrella drips should be installed. All departments and classrooms should be reached from halls and not through other rooms. Main entrances to rooms where worship is planned should be from the rear.

Provision against panic from fire should be made. At least two staircases built of fireproof material should be available from upper floors. It is not too much to require either fireproof stairs or fire escapes on all Sunday-school buildings of three stories or over.

Toilets should be conveniently located on main halls, not in dark basement corners. Those for the two sexes should not be located adjacent to each other or on the same hall.

Frescoings should be restful in character. The good colors do not cost more than those which are objectionable. Red and blue will, of course, be avoided. Soft tones of brown and green are most desirable. It is better to trust the architect and decorator in this matter than to take a vote of the committee.

Carbureted Gasoline as Fuel.—The higher calorific value of gasoline varies from 19,000 to 21,000 B.T.U. per pound, and at the combustion of each pound of fuel there is formed, on an average, 1.4 pound of water which absorbs about 1,500 heat-units for its vaporization into dry steam.

The lower calorific value is, therefore, using a rather conservative figure, 18,500 B.T.U. per pound.

The volume of one pound of gasoline vapor is, at the atmospheric pressure and 62° F.,* 4.2 cubic feet, and according to analysis there is required for its complete combustion 15 pounds, or 198 cubic feet of air per pound of the fuel.

For complete combustion in the gas-engine cylinder some margin of excess air, above what the analysis calls for, is required, and if this margin be made 15 per cent. excess there would be required, practically, $15 + 2.25 = 17.25$ pounds. The total weight of the mixture from one pound of fuel becomes, thus, 18.25 pounds.

From Table VIII of pressures of saturation of gasoline, page 106, it will appear that at the temperature of 86° F., its pressure is 251 $^m/_m$ mercury. The pressure of the atmosphere is, on an average, approximately, three times this pressure, but, on the other hand, the weight of the gasoline vapor is approximately three times as heavy as the air, and, hence, at the temperature of 86° F., a saturated mixture of air and gasoline contains equal weights of air and gas. Such a mixture is too rich in fuel to be explosive, but if the air is charged with less than one-quarter of the fuel required for saturation, and until it contains only a small percentage of fuel, it will be explosive. That a saturated mixture of air and fuel is non-explosive is illustrated by the experiment, which can be made, with due precaution, of striking fire to a

* Assuming the gasoline to be of a composition corresponding to the formula C_6H_{14}, then, according to equation 95, page 101, the density of the gas becomes $\frac{86}{28.88}$ =, approximately, three times that of air. Its volume per pound, therefore, $\frac{13.14}{3} = 4.36$ cubic feet at 62° F. For average gasoline fuels the figure 4.2 will answer, corresponding to a molecular weight of 89. Compare with experiments made by A. H. Gill and H. R. Healey, Technology Quarterly, 1902, Vol. XV, page 74.

match in a vessel containing gasoline in which the air is fully saturated with gas. Under these conditions the fuel-mixture will not burn, whereas if the match be lighted several feet away from the vessel containing the fuel, where the air may be charged with only a small suitable quantity of fuel, an explosion will result. On account of the high vapor-pressure of the fuel, and since a very small quantity of fuel is required, an explosive mixture is very readily formed, under all atmospheric conditions, if a current of air be passed over the surface of the fuel-liquid.

The effect of the vaporization of gasoline in the carbureter is to lower the temperature of the air-and-gas mixture and thereby to increase the density with which it enters the cylinder. This factor can readily be taken account of in the computation for the heating-value per unit volume of the charge.

The latent heat of gasoline is, approximately, 180 B.T.U., and there are 18.25 pounds of mixture that must supply, in the main, the heat demanded for the vaporization of each pound of fuel. Hence, each pound of the mixture must supply approximately 10 heat-units, and in doing this its temperature will be reduced approximately 40 degrees Fahrenheit; the specific heat of the air and vapor mixture being 0.25. Evidence of the material lowering of the temperature of the charge is found in the frost often forming on the carbureter.

In the general estimate for the temperature of the final charge after completed suction-stroke, it was assumed that the gas and air arrived to the engine at a temperature of 62° F. In the case of gasoline this figure must be reduced to approximately 62 − 40 = 22° F., in order to obtain the average temperature at which the carbureted gasoline-mixture is supplied.

The temperature of the final gasoline-charge after completed suction-stroke, computed as before, but starting from the temperature 22° F., instead of 62° F., will be about 86° F. The ratio, therefore, between the specific volume of the final charge after completed suction-stroke and its specific volume at standard temperature and pressure will be

$$\frac{V_a}{V_o} = \frac{P_o}{P_a} \cdot \frac{T_a}{T_o} = \frac{14.7}{13.2} \cdot \frac{546}{522} = 1.15.$$

The drop in pressure during the admission to the cylinder being allowed, as before, 1½ pound.

The temperature-changes to which the charge is subjected during its admission to the cylinder will be explained by the following schedule:

The gasoline and air arrive to the carbureter at the temperature 62° F. The temperature of the mixture after carburation = 62 − 40 = 22° F.; after expansion at entering the cylinder = 22 − 16 = 6° F.; after being heated by the cylinder = 6 + 46 = 52° F.; final charge after mingling with the neutrals = 86° F.

The estimate of the heating-value per cubic feet of suitable mixture will appear as follows:

The volume of air required by analysis per cubic foot of gasoline vapor . . a = 47.14 cubic feet.
Add 15 per cent excess air . . . . 7.07 cubic feet.
Total volume of air to be supplied per cubic foot of gasoline vapor . . . xa = 54.21 cubic feet.
The volume of the expanded normal charge containing one cubic foot of gasoline vapor

$$\frac{V_a}{V_o}(xa + 1) = 1.15 \times 55.21 = 63.49 \text{ cubic feet.}$$

Heating-value per cubic foot of gasoline vapor

$$\frac{18{,}500}{4.2} = 4{,}400 \text{ B.T.U.}$$

Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(xa + 1)} = \frac{4{,}400}{63.49} = 69.3 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression ratio to be 4, and hence the expected efficiency not less than 0.24.

$$D_1 = \frac{42.42}{Efy\dfrac{H}{\frac{V_a}{V_o}(xa + 1)}} = \frac{42.42}{0.24 \times 69.3},$$

$$\therefore D_1 = 2.55 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P. per minute, allowing 15 per cent. minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{E f y \dfrac{H}{\dfrac{V_a}{V_o}(x a + 1)}} = 1.15 \frac{42.42}{0.24 + 69.3} = 2.93 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., assuming the mechanical efficiency to be 0.85.

$$D_4 = 1.35 \frac{42.42}{0.24 \times 69.3} = 3.44 \text{ cubic feet per minute.}$$

The capacity of a gasoline-engine of given dimensions will be in B.H.P.

$$\text{B.H.P.} = \frac{l a N}{3.44 \times 3{,}456} = \frac{l a N}{11{,}900}, \text{ approximately.}$$

l being the length of stroke in square inches, a the area of the piston in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-volume is based will be found, in Table XI, to approximate 90 pounds per square inch.

The same result is approximated by the formula,

$$m.e.p. = 5.4 \times 0.24 \times 69.3 = 89.81.$$

Kerosene.—Kerosene, like gasoline, is a fuel obtained at the distillation of crude mineral oil. It is a heavier oil than gasoline; its specific gravity varying from 0.79 to 0.82, or from 47° to 41° Baumé at 62° F.

The composition of kerosene is quite closely represented by the formula $C_{10} H_{22}$, the carbon and hydrogen percentages of which are:

Carbon	0.845
Hydrogen	0.155
	1.000

Average samples of the commercial fuel-oil often analyze, approximately,

Carbon	0.845
Hydrogen	0.139
Nitrogen and Oxygen	0.016
	1.000

and they give at calorimeter tests a calorific value of 20,000 to 23,000 B.T.U. per pound.

The distillates from petroleum possess two properties that vary with their specific gravities, and which determine, largely, the arrangements by which they can be used as motor fuels. These are, the flashing-point and the fire-test or burning-point. The former is its volatility or the temperature at which the fuel gives off, when slowly heated, an ignitible vapor, and the latter is the lowest temperature at which the fuel can continue to burn. The flashing-point of kerosene is at 115° F., or thereabout and the fire-test is at approximately 140° F.

Kerosene, being less readily vaporized and having a fire-test much higher than gasoline, is a safer fuel in respect to fire-risk than gasoline, but at the same time it is somewhat less suitable for use in the gas-engine, because it requires to be heated in order to become vaporized at atmospheric pressure. It is evident that good combustion in the cylinder can be obtained only when the fuel is thoroughly vaporized at the time explosion takes place. It has been found that if the charge, at its final mixture with air or at the admission to the cylinder, becomes cooled to a temperature essentially below the flashing-point of the fuel, then the best results will not be realized at the combustion.

If the lowest temperature of the charge, allowable at its admission, be assumed to be 80° F., the temperature of the final charge at completed suction-stroke becomes approximately 160° F., and thus, at a drop in pressure of 1½ pound, the volume of the final charge becomes

$$V_a = \frac{14.7}{13.2} \cdot \frac{620}{522} V_o = 1.3\, V_o.$$

To obtain this result the fuel is often heated at its carburation

to a temperature of from 300° to 600° F., depending on the amount of air used at the carburation and on the temperature of the air later admixed at the engine.

Carbureted Kerosene Fuel.—At the combustion of kerosene there is formed 1.4 pound of water per pound of fuel, which absorbs for its vaporization 1,500 heat-units from the heat of combustion. Hence, counting the higher calorific value of the fuel as 20,000 B.T.U., the lower value becomes 18,500 B.T.U.

One pound of kerosene-vapor occupies at atmospheric pressure and at 62° F., a volume of approximately * 2.5 cubic feet, and there will be required for its combustion 190 cubic feet of air.

The estimate of the heating-value of a suitable mixture, per cubic foot, will therefore appear as follows:

The volume of air required by analysis, per cubic foot of kerosene vapor . .	$a = 76$	cubic feet.
Add 15 per cent excess air	11.4	cubic feet.
Total volume of air to be supplied, per cubic foot of vapor	$x\,a = 87.4$	cubic feet.

The volume of the expanded normal charge containing one cubic foot of vapor

$$\frac{V_a}{V_o}(x\,a + 1) = 1.3 \times 88.4 = 114.9 \text{ cubic feet.}$$

Heating-value of kerosene per cubic foot of vapor

$$H = \frac{18{,}500}{2.5} = 7{,}400 \text{ B.T.U.}$$

Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{7400}{114.9} = 64.4 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression ratio to be 4, and, hence, the expected efficiency not less than 0.24.

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* If the composition of the fuel is that expressed by the formula $C_{10}H_{22}$ then the density of the gas, according to equation 95, becomes 4.92 compared with that of air, and its volume, per pound, at 62° F. 2.64 cubic feet.

$$D_1 = \frac{42.42}{E f y \dfrac{H}{\dfrac{V_a}{V_o}(x a + 1)}} = \frac{42.42}{0.24 \times 64.4}$$

$$D_1 = 2.74 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{0.24 \times 64.4} = 3.15 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.24 \times 64.4} = 3.70 \text{ cubic feet per minute.}$$

The capacity of a kerosene-engine of given dimensions will be in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a N}{3.70 \times 3{,}456} = \frac{l\,a N}{12{,}800} \text{ approximately,}$$

l being the length of stroke, in inches; a the area of the piston, in square inches, and N the number of revolutions per minute.

By looking up in Table XI, page 120, the figure nearest to 2.74 for the maximum I.H.P., it will be noticed that this suction displacement is based on a corresponding mean effective pressure of approximately 84 pounds.

The same result is approximated by the formula

$$m.e.p. = 5.4 \times 0.24 \times 64.4 = 83.46 \text{ pounds.}$$

Properties of the Common Fuel-Gases.—Most of the fuel-gases commonly employed for motive purposes vary quite materially in composition and heating-value at different localities, and some, as for instance producer gas, even from time to time, depending on how the generator is manipulated. Any definite analysis for any particular kind of gas cannot therefore be given, or depended on. Each kind of fuel-gas has, however, its own characteristics by which each may generally be distinguished from another. In the following Table XIII, are given some sample compositions and

TABLE XIII.

Properties of Common Fuel Gases.

Name of Gas.	Constituents of the Gas, in Per Cent. Volume.							Calorific Value per Cub. Ft. at 62° F. B. T. U.		Cub. Ft. of Air Req'd. for Combustion, per c. f. of Gas, Theoretically.	Calorific Value per Cub. Ft. of Normal Charge in B. T. U. $\frac{H}{\frac{Va}{Vo}(xa+1)}$
	H.	CO.	CH_4.	C_2H_4.	O.	CO_2.	N.	High.	Low. (H)		
Blast-Furnace Gas.											
English	1.5	27.5	.....	.....	.....	6.0	65.0	95	94	0.70	42.4
Scotch	2.3	24.7	0.8	.....	.....	5.8	66.4	96	94	0.72	42.0
Swedish (Charcoal)	5.4	24.0	.....	.....	.....	12.6	58.0	95	92	0.71	41.3
United States	5.2	26.8	1.6	.....	0.2	8.2	58.0	120	114	0.92	45.0
Coke-Oven Gas	53.0	8.2	34.5	.....	.....	2.0	2.3	550	487	4.78	61.1
at Silesia, Germany	44.6	10.9	19.5	2.2	0.3	5.0	17.5	414	370	3.39	61.3
Illuminating Gas.											
London	47.9	3.7	38.6	4.5	0.3	.....	5.0	633	562	5.59	61.5
Glasgow (cannel)	39.2	7.1	40.2	10.0	0.7	0.3	2.4	719	646	6.40	62.8
Paris	52.8	5.6	32.3	5.5	0.5	.....	3.8	607	540	5.28	62.1
Carbureted Water-Gas.											
New York	30.3	26.5	24.3	15.0	0.5	1.0	2.4	674	617	5.84	65.0
Worcester	37.2	28.6	18.9	12.9	0.1	0.1	2.2	615	562	5.23	65.1
Lowe Process, San Francisco California Light and Fuel Co.	48.0	7.0	28.0	10.	0.2	2.0	4.8	624	561	5.39	63.4

Natural Gas.											
Anderson, Ind	2.01	0.73	93.07	0.47	0.42	0.26	3.02	960	862	9 06	62
Kokomo, Ind.	1.7	0.55	94.16	0.30	0.30	0.29	2.80	966	868	9.13	62
Marion, Ind.	1.4	0.60	93.57	0.15	0.55	0.30	3.42	957	860	9.04	62
Muncie, Ind.	2.5	0.4	92.67	0.25	0.35	......	3.53	954	858	9.00	61.3
Findley, Ohio	1.84	0.41	93.35	0.35	0.39	......	3.41	959	861	9.06	62
St. Mary's, Ohio	2.14	0.44	93.85	0.20	0.35	......	2.98	963	865	9.10	62
Pittsburg, Pa.	20.0	1.00	72.18	6.30	0.8	0.8	0.00	902	810	8.33	62
Producer Gas Anthracite (Suct.)											
Fair	8.2	22.0	2.4	......	......	6.4	61.0	122	116	0.96	45
Rich	12.0	27.0	1.4	......	......	2.5	57.1	141	134	1.10	48
Average	9.4	23.6	2.6	......	......	5.3	59.1	133	126	1.06	46
Producer Gas, Bituminous.											
Rotative ash table Type	13.8	20.4	3.4	......	......	9.2	53.2	145	134	1.15	50
Rotative ash table Type (Mean of 54 Tests on various coals)	12.4	19.2	3.1	......	......	9.5	55.8	134	124	1.06	45
Same producer (Average of 4 Tests on Lignite)	14.6	22.6	3.0	......	......	9.1	50.7	150	140	1.18	48
Water-bottom Type	15.3	21.5	3.6	......	......	8.5	51.0	156	144	1.23	49
Mond Gas	29.0	12.0	2.0	......	......	14.5	42.5	154	137	1.18	47
Water-Gas uncarbureted	51.8	43.4	......	......	......	3.5	1.3	310	282	2.28	63
" " "	49.5	36.0	1.0	......	......	4.2	9.3	289	262	2.06	63
" " "	30.0	28.0	......	......	......	34.0	8.0	189	174	1.39	55

heating-values of various gases that have been duly analyzed and reported on.

Natural Gas as Fuel.—Natural gas varies to some extent in respect to its calorific value, and, to be on the safe side in estimating the cylinder-capacity required for a specified number of horse-power, its heating-value should be appraised conservatively. Its low value may be assumed to be 860 B.T.U. per cubic foot of gas of standard temperature and pressure.

According to analysis there will be required for its combustion, on an average, 9.0 cubic feet of air per cubic foot of gas.

The estimate for the heating-value per cubic foot of suitable mixture becomes then:

The volume of air required by analysis per cubic foot of gas	$a =$	9.0 cubic feet.
Add 15 per cent excess air . . .		1.35 cubic feet.
Total volume of air to be supplied, per cubic foot of gas	$xa =$	10.35 cubic feet.
Total volume of the mixture, per cubic foot of gas	$xa + 1 =$	11.35 cubic feet.

The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(xa + 1) = 1.23 \times 11.35 = 13.96 \text{ cubic feet.}$$

Heating-value per cubic foot of gas . . . $H = 860$ B.T.U.

Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(xa + 1)} = \frac{860}{13.96} = 62 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression-ratio to be 5, and, hence, the expected efficiency not less than 0.26,

$$D_1 = \frac{42.42}{Efy\dfrac{H}{\frac{V_a}{V_o}(xa + 1)}} = \frac{42.42}{0.26 \times 62}$$

$$D_1 = 2.63 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{0.26 \times 62} = 3.02 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.26 \times 62} = 3.55 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions when running on natural gas will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a N}{3.55 \times 3{,}456} = \frac{l\,a N}{12{,}300},$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based is approximately 88 pounds. See Table XI, page 120.

Illuminating-Gas.—City illuminating-gas is still, occasionally, and it was not many years ago, generally, obtained by distilling off the volatile hydrocarbons from bituminous coal. The condensible products are, partly, fixed by being heated to a high temperature in the retorts, and, partly, removed at the cooling and cleaning process. The result of the process is a gas consisting mainly of hydrogen, methane and heavy hydrocarbons (illuminants). A quite average sample of coal-gas is the analysis, given by M. W. Robinson, of the Birmingham, England, city-gas, the mean composition of which approximates, by volume,

	H	CH_4	C_2H_4	CO	CO_2	air.
per cent	45	40	5	5	1	4.

At 62° Fahrenheit, the volume per pound of gas is 32 cubic feet.

The heating-value of the gas is 650 B.T.U. per cubic foot, and the volume of air required for its combustion is 5.75 cubic feet per cubic foot of gas.

In the samples of illuminating-gas given in Table XIII, the heating-value varies from 607 to 719 B.T.U. per cubic foot, and

the calorific value per cubic foot of expanded normal charge, including 15 per cent excess air, is from 61.5 to 62.8. With these figures as basis for a computation the required displacement volume of the cylinder, or the normal mean effective pressure, can readily be obtained, identically with the computation for natural gas.

Illuminating-gas is at present often produced by adding illuminants to so-called water-gas of a proper heating-value to imitate the old-fashioned retorted coal-gas. The fuel-value in this manufactured gas consists principally of hydrogen and carbon monoxide, but its actual composition varies quite materially, depending on the manipulation of the water-gas generators. The gas is somewhat lighter than coal-gas, and its heating-value varies between 550 to 600 B.T.U. per cubic foot. The air required for its combustion in the gas-engine is the same as that required by coal-gas, or between 5 to 6 cubic feet per cubic foot of gas.

Coke-Oven Gas.—Modern coke-oven plants yield considerable quantities of gas, in excess of the requirements for fuel for the coking-process. This gas is becoming more and more to be considered an important fuel for generating power in the gas-engine. The yield of gas is, however, a fluctuating factor; as it varies with the quality of the fuel, and with the different stages of the coking-process. The output of the coke-ovens, moreover, varies with the fluctuating demands from the furnaces that absorb their product, and this fact must be taken into consideration in estimating the actual value of the gas for independent industrial use.

The more recent types of regenerative coke-ovens yield, per net ton of coal, on an average 6,000 to 10,000 cubic feet of gas, respectively from coals low and from those high in volatile matter. Of this, about 40 per cent will be available as a by-product for industrial purposes—and the heating-value of the gas will vary from 500 to 700 B.T.U. per cubic foot.

The average total yield of gas may be assumed 8,000 cubic feet of a heating-value of 600 B.T.U., or 4,800,000 B.T.U. per net ton of coal. As the coking process proceeds, practically, during 24 hours, there will be obtained 200,000 B.T.U. per hour;

40 per cent of which is 80,000 B.T.U., available, on an average per hour, for industrial purposes. Utilized in an engine of an economy of, say, 10,000 B.T.U. per indicated horse-power per hour, there would, thus, be generated 8 * indicated horse-power per net ton of coal coked, which in a plant of the moderate capacity of 500 tons per day would amount to a power-reserve of 4,000 indicated horse-power, continuously.

There being connected with economical plants a process for the recovery of the by-products in the tar which is collected at the cooling of the gas, the gas will be supplied from the coking plant already partially cleaned. The tar precipitators or separators generally employed are simply high steel-plate cylinders through which the gas passes slowly up or down in a zig-zag path, baffled by shelvings, to the right and to the left, on which the tar is deposited as the gas is gradually cooled.

For a further cleaning of the gas, to make it suitable for use in the gas-engine, centrifugal cleaners, similar to those used for the cleaning of bituminous producer-gas, may be employed. The final cleaning is accomplished in so-called dry scrubbers, which are identical to those described in connection with producer-gas installations, page 436.

The composition and heating-value of the gas as well as the supply of air required for its combustion are factors that vary somewhat. For the determination of the required suction-displacement per horse-power of the engine using this gas, it may be assumed that the lower heating-value of the gas is not less than 500 B.T.U. per cubic foot and that the air required for its proper combustion, including 15 per cent excess above the theoretical requirement, is 8 cubic feet per cubic foot of gas. The heating-value of the expanded normal charge will then be 60 B.T.U. per cubic foot, or nearly at par with that of illuminating gas.

Bituminous Producer-Gas as Fuel.—The low calorific value of bituminous gas should, on an average, not be less than 138 B.T.U per cubic foot at standard temperature and pressure, and for its combustion there will be required, according to analysis, a mean

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* 10 to 12 horse-power is sometimes, less conservatively, stated.

of 1.2 cubic foot of air per cubic foot of gas. In order to obtain a complete combustion in the gas-engine cylinder it is, however, required that there should be an excess of air of in the neighborhood of 15 per cent.

Founded on these data, the estimate for the heating-value per cubic foot of suitable mixture and for the required cylinder capacity becomes:

The volume of air required by analysis per cubic foot of gas	$a = 1.2$ cubic foot.
Add 15 per cent excess air	0.18 cubic foot.
The total volume of air to be supplied per cubic foot of gas	$x\,a = 1.38$ cubic foot.
The volume of the mixture per cubic foot of gas, at standard temperature and pressure	$x\,a + 1 = 2.38$ cubic feet.

The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(x\,a + 1) = 1.23 \times 2.38 = 2.93 \text{ cubic feet.}$$

Heating-value per cubic foot of standard gas $H = 138$ B.T.U.
Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{138}{2.93} = 47 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P. per minute, assuming the compression ratio to be 6.5, and the expected efficiency, thus, not less than 0.3,

$$D_1 = \frac{42.42}{E f y \dfrac{H}{\frac{V_a}{V_o}(x\,a \times 1)}} = \frac{42.42}{0.3 \times 47}$$

$$D_1 = 3.01 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P. per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15\,\frac{42.42}{0.3 \times 47} = 3.46 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P. per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.3 \times 4.7} = 4.06 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions, when running on bituminous producer-gas, will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{4.06 \times 3{,}456} = \frac{l\,a\,N}{14{,}000}$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will be found in Table XI, page 120, opposite the values $D_1 = 3.02$, $D_2 = 3.47$ and $D_4 = 4.08$. Hence, M.E.P. = 76 pounds per square inch. The same value can be computed from the relation

$$m.e.p. = 5.4\,E\,f\,y \frac{H}{\frac{V_a}{V_o}(x\,a + 1)}$$

Anthracite Suction Producer-Gas as Fuel.—Anthracite suction gas of efficient quality has, on an average, a low calorific value of not less than 120 B.T.U. per cubic foot, and for its combustion there is required, according to analysis, 1.1 cubic foot of air per cubic foot of gas. An excess of air of about 15 per cent should, however, be provided, in order to insure complete combustion in the gas-engine cylinder.

The estimate for the heating-value per cubic foot of suitable mixture becomes:

The volume of air required by analysis per cubic foot of gas	$a = 1.1$ cubic foot.
Add 15 per cent excess air	0.165 cubic foot.
The total volume of air to be supplied per cubic foot of gas	$x\,a = 1.265$ cubic feet.
The volume of the mixture, per cubic foot at standard temperature and pressure	$x\,a + 1 = 2.265$ cubic feet.

The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(x\,a + 1) = 1.23 \times 2.265 = 2.78 \text{ cubic feet.}$$

Heating-value per cubic foot of standard gas $H = 120$ B.T.U.
Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{120}{2.78} = 43.2 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P. per minute, assuming the compression ratio to be 6.5 and the expected efficiency, thus, not less than 0.3:

$$D_1 = \frac{42.42}{E\,f\,y\,\frac{H}{\frac{V_a}{V_o}(x\,a + 1)}} = \frac{42.42}{0.3 \times 43.2}$$

$$D_1 = 3.27 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity:

$$D_2 = 1.15\,\frac{42.42}{0.3 \times 43.2} = 3.76 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P. per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35\,\frac{42.42}{0.3 \times 43.2} = 4.41 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions when running on suction gas will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{4.41 \times 34{,}56} = \frac{l\,a\,N}{15{,}240}, \text{ approximately.}$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will be found in Table XI, page 120. It

approximates the value opposite $D = 3.27$ or 70 pounds per square inch.

The same result will be approximated by solving equation 50*b*.

Thus, $m.e.p. = 5.4 \times 0.3 \times 43.2 = 69.98$ pounds.

Comparing the figures for the suction-displacement and the mean effective pressure obtained for anthracite suction gas with those obtained for bituminous gas, it will be seen, that they vary in the same ratio as the heating-value per cubic foot of normal expanded charge. That this must be so is evident, because, other things being equal, in the rate that more heat-units are put into an engine for transformation in that rate its capacity for doing work must increase.

The Engine-Power at an Elevation above the Sea-Level. The normal suction-displacement per horse-power and the mean effective pressure that may be derived from the various gas-engine fuels has in the preceding paragraphs been determined on the basis that the charge is supplied to the engine under the pressure of the atmosphere at the sea-level (14.7 pounds). The fact that the gas alone, perhaps, is supplied at a pressure somewhat higher, or lower, than the atmosphere will not materially affect the power of an engine, as long as the air for the charge is supplied under the pressure of the free atmosphere.

If an engine were to work under an atmospheric pressure less than 14.7 pounds, as would be the case if installed at a high altitude, then this fact must be taken into consideration in determining its power.

The heating-value of the expanded normal charge at sea-level is

$$\frac{H}{\frac{V_a}{V_o}(xa+1)} \quad \text{or} \quad \frac{H}{\frac{p_o}{p_a}\frac{T_a}{T_o}(xa+1)}.$$

For an atmospheric pressure 14.7 pounds, p_a has been assumed to be ($p_o - 1\frac{1}{2}$) 13.2 pounds, or a total drop in pressure between the outside and the pressure in the cylinder at the end of the suction-stroke of $1\frac{1}{2}$ pound has been allowed. This normal drop in pressure would presumably become proportionately less as the atmosphere becomes rarefied at altitudes elevated above the sea-level.

Thus, the pressure in the cylinder at the higher altitude

$$p_a' = \frac{p_o'}{p_o}(p_o - 1\tfrac{1}{2}),$$

when p_o' is the local atmospheric pressure, in pounds.

Or,
$$p_a' = \frac{B_o'}{B_o}(p_o - 1\tfrac{1}{2}),$$

when B_o' is the local barometric pressure and B_o the barometric pressure at the sea-level. The heating-value of the expanded normal charge under the barometric pressure B_o' becomes

$$\frac{B_o'}{B_o} \frac{H}{\frac{p}{p_a}\frac{T_a}{T_o}(x\,a + 1)},$$

or it becomes reduced in the ratio $\frac{B_o'}{B_o}$.

As the heating-value of the expanded normal charge determines the mean effective pressure, and the horse-power of an engine, therefore these quantities will, under a barometric pressure B_o', also be reduced in the ratio $\frac{B_o'}{B_o}$.

The horse-power, $H.P._{el}$, at an elevation compared with the horse-power at the sea-level, $H.P_{sl}$, becomes, thus

$$H.P._{el} = \frac{B_o'}{B_o} H.P._{sl}.$$

Example.—What will be the normal rated brake horse-power of a 16 × 24 four-cycle engine, 200 revolutions per minute, working on anthracite producer-gas at an altitude of 4,000 feet?

The normal barometric pressure at an elevation of 4,000 feet is 26″ inches mercury.

thus,
$$H.P._{4000} = \frac{26}{30} H.P._{sl}.$$

The 16 × 24 engine at the sea-level is, according to Table XXV, page 274, of 62 rated brake horse-power. Hence at an elevation of 4,000 feet it is $\frac{26}{30} \times 62 = 0.87 \times 62 = 54$ rated brake horse-power.

Blast-Furnace Gas.—The, so-called, waste gas from blast-furnaces has for years been an important fuel in connection with the operation of the furnaces. Of greatest importance is its use for the heating of the hot-blast stoves and for generating steam by which blowing-engines and dynamos are operated. For the latter purpose, however, the gas has proven not very efficient, and is at times even too lean to burn satisfactorily. Of late years, the gas has been extensively used in large gas-engines, for generating abundantly the motive power required for blast-engines and auxiliaries and by this system of utilizing the gas there has been accomplished a material saving in expense for additional fuel.

At the production of one ton of pig-iron there is used, on an average, one ton of coke; and, approximately, six tons* of gas are generated. Estimates place the requirement of gas for the hot-blast stoves, and other heating purposes, at about 30 per cent of the total output, or at about 3,600 pounds per ton of pig; the remaining 8,400 pounds of gas will, therefore, be available for power purposes.

The analysis of an average sample of the gas would show it to be of a heating-value of approximately 1,320 B.T.U. per pound, or a total of 11,088,000 B.T.U. in 8,400 pounds. Assuming an engine utilizing this gas to have a thermal efficiency of 0.24, which is common with respect to large engines, there would, then in the way of power, be obtained, per ton of pig-iron produced per hour, $\frac{11{,}088{,}000 \times 0.24}{2{,}545} = 1{,}040$ indicated horse-power, or approximately 884 † brake horse-power.

The power required for driving blowing-engines, pumps, etc., may be liberally figured at 284 horse-power per ton of pig-iron produced per hour; hence there will remain approximately 600 horse-power, in excess of the requirements for the furnace.

The gas is, after leaving the top of the blast-furnace, carried through the down-comer into a system of dust-catchers, which

* Corresponding to 158,000 cubic feet per ton of pig. The volume is variously estimated at from 180,000 to 120,000 cubic feet, varying with the class of ore and with the amount of coal consumed.

† Variously estimated at from 750 to 1,200 B.H.P.

collect the greater part of the dust with which the gas is charged, and it is in this manner made clean enough for use as fuel. For being utilized in the gas-engine it is, however, required that the gas be further cleaned, to eliminate practically all the fine dust, which, otherwise, will prove injurious to the engine.

Fig. 36 is a diagrammatic view of a washing-plant for blast-furnace gas.

After having deposited a large part of its dust in the dust-collector, the gas is led into the tower washer, which consists of three high cylinders containing, each, several grids over which a shower of water is sprayed from spray pipes above. The gas is passed up through one after the other of these cylinders, as plainly indicated by the small arrows in the figure, and when leaving the apparatus its dust-contents has become reduced to 1.5 to 1.0 gramme per cubic metre * of gas.

From the primary washer the gas is passed into the centrifugal cleaner, which in the figure is a Thiesen washer.

This apparatus consists of a casing containing a revolving drum which is provided with helical vanes on its outside surface. Between the drum, which revolves at a speed of 350 revolutions per minute, and the casing the gas is passed and is vigorously thrown out against a film of water passing over the inside surface of the casing; the water entering through valves at *a* and drained off, together with the dust contents, into a drain and seal basin at *b*.

The gas is drawn in to the washer by means of a fan driven by the drum shaft and located in the casing at *c*; the gas-current is retarded by the helical drum-vanes, and then again given a suitable discharge-speed by a fan located at the exit end of the casing, at *d*.

An apparatus of this type, cleaning 720,000 to 1,000,000 cubic feet of gas per hour, requires a motor of about 150 horse-power; the power-requirements being figured generally 0.15 to 0.2 horse-power per 1,000 cubic feet of gas. The gas leaving the washer is customarily tested for dust, at intervals, and the quantity runs normally from 0.01 to 0.02 gramme per cubic metre of gas.

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* Equivalent, practically, to 3 to 2 grains per cubic foot.

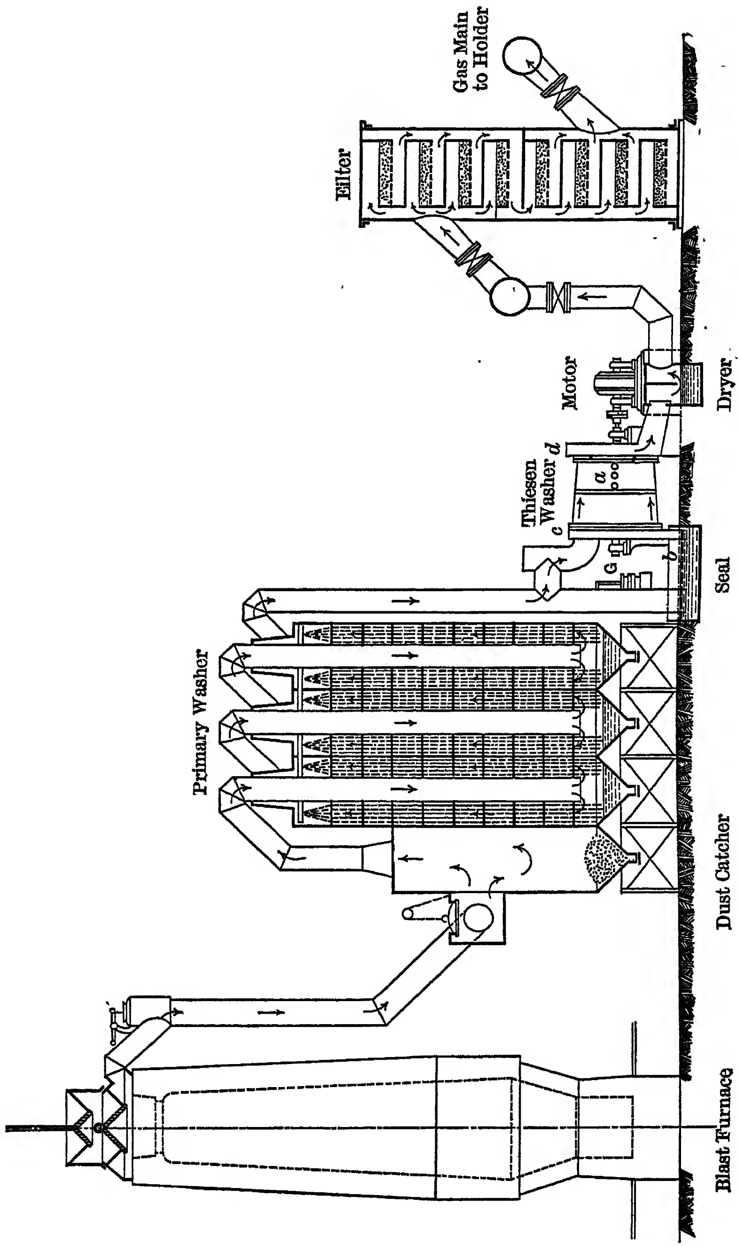


FIG. 36.—Washing Plant for Blast-Furnace Gas.

From the washer the gas is conducted, through a dryer, to a filtering-tower, which consists of a cylinder containing a double set of shelves charged with filtering material through which the gas passes, in the manner plainly indicated by the small arrows in the figure, and from there it is delivered to the gas-holder and the engine.

At G is shown a gasometer, which has for purpose to shut down the Thiesen washer the moment the pressure in the delivery pipe from the primary washer becomes dangerously close to that of the atmosphere. Without such an apparatus the plant would be unsafe, as, with the washer continuing to draw from the primary washer after the gas might have been shut off from the furnace, air would be drawn in to the gas-system, at the risk of producing an explosive mixture in the holder and gas-main.

The gas used in the hot-blast stoves and boilers is, according to the latest practice, passed from the dust-catcher also through a simple stationary washer, to prevent the choking of gas-flues by dust, which otherwise may occur.

The composition of blast-furnace gas varies considerably, but it contains, under normal conditions, only a small percentage of hydrogen, which makes it suitable for a high compression in the gas-engine. At times, however, a very large percentage of hydrogen may be present, and this fact must be taken account of, when deciding upon the proper compression to be carried by the engine. While a compression-pressure of 190 pounds has been frequently used for this gas, it has, due to difficulties, recently been cut down considerably. A pressure of approximately 160 pounds should be considered safe.

The following is a normal composition of blast-furnace gas:

	Per Cent Volume	Per Cent Weight
CO	27.0	26.32
H	2.5	0.18
CH_4	0.5	0.29
CO_2	9.0	13.79
N	61.0	59.42
	100.0	100.00

Heating-value per cubic foot at 62° F., and 1 atmosphere 102 B.T.U.

Volume per pound of gas at 62° F. and 1 atmosphere 13.2 cubic feet.

For complete combustion of one pound of gas there is required 0.76 pound of air; or, generally, there is required per cubic foot of gas approximately, ¾ cubic foot of air.

Including 15 per cent excess air, there should be supplied for combustion in the gas-engine 0.86 cubic foot of air per cubic foot of gas.

CHAPTER VII

ALCOHOL FUELS

IN France and Germany there has for a number of years been made a considerable effort to encourage the use of alcohol as fuel for motor purposes, and to replace, as much as possible, gasoline (an imported commodity) by alcohol. In the United States there has also developed, lately, a purpose to extend the markets for fuel-supply beyond the control of a mining-monopoly, and to open them for the modern fuel, in the interests of the agriculture of the nation.

The ample investigations made for the solution of the proposition make it appear certain, at present, that, as an engineering proposition, the problem is solved, and it remains an open question only so far as its proper economy is concerned.

The Alcohols.—The alcohols manufactured are of two kinds: ethyl alcohol, $C_2 H_6 O$, obtained through fermentation and distillation of sugar or starch; and methyl alcohol, $C H_4 O$, obtained by destructive distillation of wood.

The following table contains some of the principal data regarding these products in their pure state.

Product.	Specific Gravity.		Specific Heat.		Boiling Point at 760 m/m Degrees Fahr.	Latent Heat, in Vapor.		Vapor Tension. Lbs. per sq. in.
	At 32° F.	At 62° F.	Of Gas at const. press.	Of Liquid.		From 62° F.	Above Boiling Point.	
Ethyl Alcohol . $C_2 H_6 O$	0.806	0.794	0.453	0.726	173	440	365	0.7
Methyl Alcohol $C H_4 O$	0.810	0.798	0.458	0.670	148	510	475	1.5

For the complete combustion of the alcohols there is required:

	Oxygen per pound of fuel	Air per pound of fuel
by C_2H_6O	2.086 pounds	9.03 pounds
by CH_4O	1.5 pounds	6.5 pounds

Compared with gasoline, the alcohols are of rather low heating-value; when pure, their calorific power compares with that of the former fuel as follows:

Gasoline	20,000 B.T.U.
Ethyl alcohol	16,000 B.T.U.
Methyl alcohol	13,000 B.T.U.

The manufactured alcohols are never pure, however, but hydrated to the extent of containing from 7 to 50 per cent water, and their actual heating-values, as fuels, will be on an average only:

for Ethyl alcohol (water contents 14%) 13,700 B.T.U.
for Methyl alcohol (water contents 14%) 11,000 B.T.U.

One requirement, in order to make it practical to allow the free production and sale of alcohol fuels on a large scale, was to find a suitable denaturant that would make the product unusable for other purposes than as fuel. There prevail, for the purpose, two satisfactory methods, the results of which are generally termed, respectively, "denatured alcohol" and "carbureted alcohol."

The composition of the former mixture is defined by law in the United States and in France; it being required that: "To 100 volumes ethyl or grain alcohol of a strength of not less than 90 per cent there must be added 10 volumes methyl or wood alcohol and ½ volume of a heavy hydrocarbon, pyridine—denaturation benzol."

The latter ingredient, required to have the boiling-points between the temperatures 280° and 390° F., is added for the purpose only of giving to the fuel a distinctive color and odor that can readily be recognized at inspection.

Carbureted alcohol is a mixture of any optional proportion

of the denatured alcohol with benzol; the latter being a purer article than that used for denaturing purposes.

Benzol and pyridine are products from by-product coke-ovens. They are increasingly used abroad for the enriching and denaturing of alcohol for motor purposes, and as illuminants in manufactured gas. Outside of their uses for these purposes, they are valuable raw materials in manufacturing chemicals. Commercial 90 degrees benzol, of a specific gravity of 0.88 at 62° F. and boiling between the temperatures 200° and 300° F., is of about 18,000 B.T.U. per pound, or of a heating-value a little less than that of gasoline.

A mixture of equal parts of denatured alcohol and benzol has been extensively tried for motor use, and in France a series of competitive tests have been made with this fuel and with pure denatured alcohol to ascertain their relative advantages *

The analyses of the two fuels were as follows:

	Denatured Alcohol	Carbureted Alcohol 50 per cent Benzol
Carbon, C	0.4372	0.6899
Hydrogen, H	0.1112	0.0948
Oxygen, O	0.3029	0.1457
Water, H_2O	0.1408	0.0685
	0.9921	0.9989
Computed higher heating-value	9,938 B.T.U.	14,225 B.T.U.
Heating-value by test	10,630 B.T.U.	14,180 B.T.U.

Some of the results obtained at the tests were:

Fuel Consumption; Ratio by Weight

Power Developed B.H.P.	Denatured Alcohol	50% Carbureted Alcohol
8.3	10	7.66
16.3	10	6.85
34.4	10	7.61

The conclusions were, that, for the same power, the consumption of 50 per cent carbureted alcohol will average $\frac{7}{10}$ of that of denatured alcohol.

* Concours International De Moteurs et Appareils utilisant L'Alcool dénaturé 1902.

Numerous competitive tests have also been made between denatured alcohol and gasoline, the results of which tend to show that, on an average, the consumption of denatured alcohol will be 1.05 pound per brake horse-power per hour against a consumption of 0.7 pound of gasoline, or, for equal power, in the ratio of 1½ pound of alcohol to 1 pound of gasoline. The latter ratio expressed in volumes would be 1.38 gallon of alcohol to 1 gallon of gasoline. Counting the lower heating-value of gasoline 18,500 and that of denatured alcohol 10,000 B.T.U., the efficiency of the two fuels becomes in the ratio of 24 for gasoline to 30 for alcohol.

Comparisons between the two fuels have been stated in different ways, and the figures will vary, to some extent, with the type of engine to which they have reference. For instance, during the sessions of the Motor Union of Great Britain and Ireland, which had for purpose to investigate the relative values, as motor-fuels, of gasoline and alcohol, it was stated: "It has been brought out through evidence that petrol (gasoline), and alcohol stand in the ratio of 2 to 1 as regards their heat of combustion, but that in the case of alcohol 30 per cent of the heat is available, while in the case of petrol 20 per cent can be obtained. We require then 4 parts of alcohol to 3 of petrol, by weight."

The Elementary Components of Alcohol Fuels.—The specific gravities of the two fuel-alcohols being so nearly the same, and varying somewhat with the purity of the products, any distinction between these quantities will be of no purpose in any ordinary computation. It will be assumed in the following that both are 0.8 at 62° F.

The specific gravity of benzol is 0.88 and that of water at 62° F. may be called 1.0.

Example.—Denatured alcohol consists of:

90 volumes of ethyl alcohol, C_2H_6O
10 volumes of water, H_2O
10 volumes of methyl alcohol, CH_4O
0.5 volumes of benzol, C_6H_6.

What are the weight percentages of the elementary components?

The following scheme for a dissecting analysis may conveniently be used:

Specific Weight.	Ratio of Volumes of Components.	Ratio of Weights of Components.	Weight percentages of Components.	Symbol for Components.	Mol. Weights of Components.	Ratio Between Weights of Elements.			Percentages of Weights of Elements.		
						C.	H.	O.	C.	H.	O.
1	2	3	4	5	6	7	8	9	10	11	12
0.8	90	72	0.7961	C_2H_6O 24 6 16	46	$\frac{24}{46}$	$\frac{6}{46}$	$\frac{16}{46}$	0.4154	0.1038	0.2770
1.0	10	10	0.1106	H_2O 2 16	18	...	$\frac{2}{18}$	$\frac{16}{18}$	......	0.0123	0.0984
0.8	10	8	0.0885	CH_4O 12 4 16	32	$\frac{12}{32}$	$\frac{4}{32}$	$\frac{16}{32}$	0.0333	0.0111	0.0444
0.88	0.5	0.44	0.0048	C_6H_6 72 6	78	$\frac{72}{78}$	$\frac{6}{78}$	.....	0.0044	0.00037	......
	110.5	90.44	1.0000						0.4531	0.12757	0.4198

How the figures in the above table have been derived is evident. It may be necessary only to state that the figures of the last three columns are the products of the weights of elements, columns 7, 8 and 9, and corresponding weight percentages of components, column 4.

The percentages of the elements are:

$$C = 0.453$$
$$H = 0.127$$
$$O = 0.420$$

The water contents having no influence in respect to the heating-value of a fuel, it may be desired to keep it separate.

The percentages in that case will be:

$$C = 0.453$$
$$H = 0.115$$
$$O = 0.321$$
$$H_2O = 0.111$$

The specific weight of denatured alcohol will be $\frac{90.44}{110.5} =$ 0.8185 or, approximately, 0.818 at 62° F.

Assuming that a carbureted alcohol-fuel consists of equal

volumes of denatured alcohol and benzol, the weight percentages of the elements of the mixture will be:

Components.	Specific Weight.	Weight percentages of Components.	Ratio of Weights of Elements.				Percentages of Weights of Elements.			
			$C.$	$H.$	$O.$	$H_2O.$	$C.$	$H.$	$O.$	$H_2O.$
Denatured alcohol	0.818	0.4817	0.453	0.115	0.321	0.111	0.21821	0.05540	0.15463	0.05346
Benzol	0.88	0.5183	$\frac{72}{78}$	$\frac{6}{78}$	.....	.....	0.47833	0.04000	....	....
	1.698	1.0000					0.69655	0.09540	0.15463	0.05346

The weight-percentages of the elements, thus,

$$C = 0.697$$
$$H = 0.095$$
$$O = 0.155$$
$$H_2O = 0.053$$

The weight-percentages of the components of 50 per cent carbureted alcohol are:

Components.	Ratio of Volumes.	Specific Weights.	Ratio of Weights.	Percentage Weight.
Denatured alcohol	0.5	0.818	0.409	0.4816
Benzol	0.5	0.88	0.44	0.5184
			0.849	1.0000

In France and in the United States the strength of alcohol is uniformly expressed by stating what percentage *volume* there is of pure alcohol in a diluted mixture. In Germany, however, its strength is always expressed by the percentage *weight* the pure alcohol is of the whole. In comparing results, therefore, reported from the different countries, this difference should be allowed for. For instance, an alcohol called in this country 90 per cent will be in Germany 0.858 per cent. For the reduction from per cent volume to per cent weight, or vice versa, a table of experimental determinations of the density of different hydrations will be necessary. In the Smithsonian table, below, for ethyl alcohol, the densities are given at 60° F., and it should be observed that

for each degree the temperature of the fluid is higher or lower than 60° its density will be approximately 0.0005 less or more than that given in the table.

Specific Gravity of Ethyl Alcohol Diluted by Various Percentages of Water.

Specific Gravity at 60° F. compared with Water at 60° F.	Percentage of Alcohol.		Specific Gravity at 60° F. compared with Water at 60° F.	Percentage of Alcohol.	
	By Weight.	By Volume.		By Weight.	By Volume.
0.834	85.8	90.0	0.822	90.4	93.4
.833	86.2	90.3	.821	90.8	93.7
.832	86.6	90.6	.820	91.1	94.0
.831	87.0	90.9	.819	91.5	94.2
.830	87.4	91.2	.818	91.9	94.5
.829	87.7	91.5	.817	92.2	94.8
.828	88 1	91.8	.816	92.6	95.0
.827	88.5	92.1	815	93.0	95.3
.826	88.9	92 3	.814	93.3	95.5
.825	89.3	92.6	.813	93.7	95.8
.824	89.6	92.9	812	94.0	96.0
.823	90.0	93.2	.....	.	....

In the custom house alcohol is given in "proof" gallons, that is, in gallons containing 50 per cent alcohol, by volume, 50 per cent being water. When a quantity of alcohol is given in proof gallons it is, therefore, expressed by a figure twice as large as it would be if the measure were given in gallons of 100 per cent alcohol.

Specific Heat of the Fuel-Vapors.—Denatured alcohol consists of:

Ethyl alcohol	C_2H_6O	0.796 per cent weight.
Methyl alcohol	CH_4O	0.088 " " "
Benzol	C_6H_6	0.005 " " "
Water	H_2O	0.111 " " "
		1.000

50 per cent carbureted alcohol consists of:

Ethyl alcohol	C_2H_6O	0.383 per cent weight.
Methyl alcohol	CH_4O	0.042 " " "
Benzol	C_6H_6	0.521 " " "
Water	H_2O	0.054 " " "
		1.000

Based on these compositions of the two fuel-alcohols we obtain:

The specific heat of denatured alcohol-vapor.

C_2H_6O	$0.796\times0.453=0.360$
CH_4O	$0.088\times0.458=0.0403$
C_6H_6	$0.005\times0.3=0.0015$
H_2O	$0.111\times0.48=0.0532$
	Total 0.455

The specific heat of 50 per cent carbureted alcohol-vapor.

C_2H_6O	$0.383\times0.453=0.173$
CH_4O	$0.042\times0.458=0.019$
C_6H_6	$0.521\times0.3=0.156$
H_2O	$0.054\times0.48=0.026$
	Total 0.374

Air Required for Combustion.—In the following table are given the oxygen and air required for the complete combustion of the alcohol-fuels and components, and the combustion-products obtained per pound of fuel:

Fuel.	Oxygen Required for Combustion. Lbs.	Air Required for Combustion. Lbs.	Combustion-Products.		
			CO_2. Lbs.	*H_2O.* Lbs.	*N.* Lbs.
Ethyl alcohol, C_2H_6O ..	2.087	9.037	1.913	1.174	6.95
Methyl alcohol, CH_4O ..	1.5	6.50	1.375	1.125	5.00
Benzol, C_6H_6	3.1	14.32	3.385	0.692	10.243
Denatured alcohol	1.809	7.833	1.661	1.149	6.024
50 % Carbureted alcohol	2.448	10.600	2.530	0.918	8.160

The higher heating-values of the two alcohol-fuels are:

	Computed Value	Value by Test
Heating-value of denatured alcohol	11220	11880
Heating-value of 50% carbureted alcohol (50% alcohol) . . .	14830	14200
Heating-value of 80% carbureted alcohol (80% alcohol) . . .	12050	

At complete combustion of alcohol there is formed carbon dioxide and water, as:

$$C_2H_6O + 6O = 2CO_2 + 3H_2O.$$

When the combustion is incomplete, however, there is formed aldehyde and water, as:

$$C_2H_6O + O = C_2H_4O + H_2O,$$

or acetic acid and water, as:

$$C_2H_6O + 2\,O = C_2H_4O_2 + H_2O.$$

Vapor Pressure and Critical Temperature of an Explosive Gas Mixture.—The following table, according to Sorel, gives the vapor-pressure of saturation at various temperatures for the alcohol-fuels commonly used, or it may be said to give the critical temperatures for various pressures.

TABLE XIV.

Vapor Pressure of Saturation for Denatured and 50 Per Cent Carbureted Alcohol, in Millimeters of Mercury.

Temperature.		Denatured Alcohol.	Carbureted Alcohol.	Temperature.		Denatured Alcohol.	Carbureted Alcohol.	Temperature.		Denatured Alcohol.	Carbureted Alcohol.
°C.	°F.			°C.	°F.			°C.	°F.		
−14	6.8	....	19.5	0	32	15	43	14	57.2	35	82.5
−13	8.6	....	20.5	1	33.8	16	45	15	59	37	86
−12	10.4	....	22	2	35.6	17	47.5	16	60.8	39.5	90
−11	12.2	....	23	3	37.4	18	50	17	62.6	42	93.5
−10	14	....	24.5	4	39.2	19	52.5	18	64.4	45	97.5
− 9	15.8	....	26	5	41	20	55	19	66.2	48	102
− 8	17.6	....	27.5	6	42.8	21.5	57.5	20	68	51	106.5
− 7	19.4	....	29	7	44.6	23	60.5	21	69.8	54	111.5
− 6	21.2	11.5	31	8	46.4	24.5	63	22	71.6	57.5	117.5
− 5	23	12	32.5	9	48.2	26	66.5	23	73.4	61	123.5
− 4	24.8	12.5	35.	10	50	27.5	70	24	75.2	64	130
− 3	26.6	13	37	11	51.8	29	72.5	25	77	68	136.5
− 2	28.4	13.5	38.5	12	53.6	31	75.5	26	78.8	72	145
− 1	30.2	14.5	41	13	55.4	33	79	27	80.6	76.5	.....

With regard to the alcohol-fuels, it becomes of special interest to find the minimum temperature at which a proper explosive mixture can be formed. This temperature may readily be determined, on the basis of the preceding table, when it is remembered that the ratio of the pressures of, respectively, the air and

fuel-vapor in a saturated mixture is equal to the inverted ratio of the volumes of the two gases present (see page 107).

Let B be the barometric pressure to which a mixture of fuel-vapor and air is subjected,

t and $(a + t)$ its temperature, respectively, above O° F., and above the absolute zero,

d the density of the vapor at 32° F., and at atmospheric pressure,

ϵ the coefficient of expansion by heat of air and of the vapor,

V the volume of the air mixed with each pound of the fuel, at 32° F., and atmospheric pressure, in cubic feet,

x the vapor-pressure of the fuel,

and, thus, $B - x$ the vapor-pressure of the air in the mixture.

The volume of one pound of vapor, at the temperature t and barometric pressure B, becomes, then, according to equation 11

$$\frac{760}{B} \frac{1}{d} \epsilon (a + t).$$

and the volume of the corresponding proportion of the air is

$$\frac{760}{B} V \epsilon (a + t).$$

Hence,
$$d V = \frac{x}{B - x}, \text{ or}$$

$$x = \frac{B}{1 + d V}.$$

If the fuel-and-air mixture is subjected to the atmospheric pressure, then the pressure of the vapor becomes

$$x_a = \frac{760}{1 + d V},$$

or when x_a is a known quantity we may say

$$x = \frac{B}{760} x_a.$$

For each fuel-vapor there is required a certain proportion of air for its complete combustion in a gas-engine cylinder. Practice has shown that gasoline requires for effective combustion, approximately, 15 per cent more air than the quantity theoretically

necessary, and, though it has not as yet been fully settled, there is no doubt but that alcohol-fuels should be supplied with an excess of air of about 50 per cent above that required by analysis. Under these conditions, the best mixture for the following fuels would be:

Gasoline, 1 pound to air 17.3 pounds, against 15 pounds by analysis. Denatured alcohol, 1 pound to air 11.7 pounds, against 7.8 pounds by analysis. 50 per cent carbureted alcohol, 1 pound to air 15.9 pounds, against 10.6 pounds by analysis.

In the following table are computed the vapor-pressure of some fuels in mixture with such proportions of air as will be theoretically, or practically, required for complete combustion:

TABLE XIV *A*

FUEL.	Proportion Air.	*V*.	*d*.	x m/m	Critical Temp'rature Degrees F.
Gasoline	Am't. req'd. by a'ly's	186	0.24	16.6	....
Gasoline	1.15× " " " "	214 0	.....	14.5	....
Ethyl alcohol, pure	" " " "	112.7	0 129	49.2	+72
Methyl alcohol, pure	" " " "	80.6	0.089	93.0	68
Denatured alcohol 90%	" " " "	96.7	0.101	70.6	78
Denatured alcohol, 90%	1.50× " " " "	145.1	.....	48.6	67
50% Carbureted alcohol	" " " "	131.4	0.162	34.1	24
50% Carbureted alcohol	1.50× " " " "	197.2	.....	23.0	12

In the last column is given the minimum temperature at which the mixture can exist, at atmospheric pressure, without the fuel becoming condensed. These temperatures are taken from Tables VIII and XIV.

The preceding table shows that 48.6 $^{m}/_{m}$ mercury is the vapor-pressure of the fuel in a suitable mixture of denatured alcohol and air, and 67° F. is the minimum temperature at which such a mixture can exist and still retain the full charge of the fuel in vapor, under 760 $^{m}/_{m}$ barometric pressure. It also shows that the vapor-pressure of a 50 per cent carbureted alcohol-fuel in a suitable mixture, and at 760 $^{m}/_{m}$ barometric pressure, is 23.0 $^{m}/_{m}$ mercury, and the minimum temperature at which it

can exist is 12° F. The air supply in both cases being figured 50 per cent in excess of that required by analysis.

When a fuel-mixture is of a partial vacuum, as it will be during the suction stroke in an engine-cylinder, then its vapor-pressure and the corresponding critical temperature become lower. If it be assumed that the minimum suction-pressure is 13.2 pounds, or 684 $^m/_m$ mercury, then the vapor-pressure of denatured and of 50 per cent carbureted alcohol, in a mixture including 50 per cent excess air, becomes, respectively,

$$\frac{13.2}{14.7} \times 48.6 = 43.7\ ^m/_m \text{ and } \frac{13.2}{14.7} \times 23 = 20.7\ ^m/_m,$$

and the corresponding critical temperatures will be 63° and 9° F.

The vapor-pressure of a mixture not including any excess air would, in other respects under the same conditions, be for denatured alcohol 63.5 $^m/_m$ and for 50 per cent carbureted alcohol 30.7 $^m/_m$, and the corresponding critical temperatures, respectively, 75° and 18° F.

The Minimum Initial Temperature of the Air and Alcohol Charge.—If it should be required that the fuel were completely vaporized in the carbureter before being admitted to the cylinder, then the latent heat of vaporization may be assumed to be all supplied from the sensible heat of the fuel and air, which would practically be the case if the vaporization were to take place in a common unheated carbureter.

The latent heat of vaporization from 62° F. of denatured alcohol is approximately 525 B.T.U. per pound, and that of 50 per cent carbureted alcohol is 350 B.T.U. That the latent heat of the former fuel is considerably greater than that of the latter is due to the greater percentage of water present, whose latent heat is high.

The specific heat of the gases from the two fuels may be assumed, according to the estimate, page 155, to be:

That of denatured alcohol-gas = 0.455
That of carbureted alcohol-gas = 0.374
and that of air is approximately = 0.24.

Assuming the allowance of air per pound of fuel to be:

	Air theoretically required.	Air including 50% excess.
For denatured alcohol .	7.8	11.7
For carbureted alcohol	10.6	15.9
The specific heat of the gas-and-air mixture becomes then,		
For denatured alcohol .	$\frac{0.24\times7.8+0.455}{8.8}=\frac{2.327}{8.8}$	$\frac{0.24\times11.7+0.455}{12.7}=\frac{3.263}{12.7}$
For carbureted alcohol .	$\frac{0.24\times10.6+0.374}{11.6}=\frac{2.918}{11.6}$	$\frac{0.24\times15.9+0.374}{16.9}=\frac{4.19}{16.9}$
And the fall in the temperature of the mixture, due to the abstraction of heat for the vaporization of the gas,		
For denatured alcohol	$\frac{525}{2.327}=226°F.$	$\frac{525}{3.263}=161°F.$
For carbureted alcohol .	$\frac{350}{2.918}=120°F.$	$\frac{350}{4.19}=83°F.$
Hence, to effect a complete vaporization of the fuel, the air and fuel should be initially of a temperature of, at least,		
For denatured alcohol .	226+75=301°F.	161+63=224°F.
For carbureted alcohol .	120+18=138°F.	83+9 = 92°F.

By early engine-trials it was found that alcohol-fuels could not be utilized without pre-heating the charge considerably, and in many cases it was found impossible to start an engine on a cold alcohol-charge; therefore, in such cases, gasoline was resorted to as a means for starting and heating up the alcohol-engine. It appears also evident from the preceding estimate of the critical temperatures, that, in order to vaporize an alcohol-fuel (particularly denatured alcohol) completely in the carbureter, it is required that the air be highly pre-heated.

It will be practical, however, to vaporize the fuel, only partially, in the carbureter and to allow complete vaporization to take place when, during the suction-stroke, the new charge is

mixed with the hot neutrals from the preceding charge. By late experiments it has been found that, while some pre-heating is favorable for complete combustion and high efficiency, it will not be necessary, or desirable, to heat the air to the temperature that a complete vaporization of the fuel in the carbureter would call for.

In some instances the pre-heating of the air to only some 150° F., though the compression used was quite moderate, gave rise to self-ignitions, which, of course, on account of the uncertainty in regard to the exact point of the stroke where ignition may occur, must be considered objectionable. With a properly constructed and cooled combustion-chamber any difficulty on this score should, however, readily be avoided.

To make it possible to employ a very high compression and obtain high efficiency, some builders inject water to the cylinder with the fuel. This is a feature of the Banki, the Deutz and the Marienfelde oil and alcohol engines, tests of which are recorded in Table XXXI, page 410.

The alcohols vaporizing much more slowly than gasoline, it is, with respect to them, of greater necessity to effect a thorough mixture between the air and fuel in the carbureter, and to provide ample port-areas, so as to insure a slow fluid velocity through the carbureter and intake-chambers, thereby avoiding any considerable pressure-reduction in the cylinder at the suction-stroke.

Denatured Alcohol as Fuel.—Practice showing that any high pre-heating of the charge is undesirable, it may be assumed, for the sake of an estimate of the required cylinder capacity of an alcohol-engine, that the vaporization of the fuel is completed first at the time of the beginning of the compression-stroke, and that the temperature of the charge, at that time, is consistently low.

Carrying out, then, the estimate with reference to a denatured alcohol-fuel, the temperature of the charge at the beginning of the compression should not be less than 63° F., at a pressure of $684^{m}/_{m}$ mercury, and the volume of the expanded charge per pound of fuel becomes

$$\frac{V_a}{V_o}(x\,a + 1) = \frac{760}{684}\,\frac{523}{522} \quad (x\,a + 1) = 1.1\,(x\,a + 1).$$

Assuming that the higher calorific power of a denatured alcohol

fuel is 11,700 B.T.U., its lower calorific value, after deducting the latent heat in about 1.15 pound of water-vapor formed at its combustion, will be, approximately, 10,450 B.T.U. The air necessary for its complete combustion is, theoretically, 7.8 pounds per pound of fuel, or 102.5 cubic feet at 62° F., and atmospheric pressure.

Each pound of fuel, in vapor, at this temperature and pressure occupying 9.4 cubic feet, we get:

The volume of air necessary according to analyses, per cubic foot of fuel-vapor . . . $a = 10.9$ cubic feet.

Add 50 per cent excess air . . . . . $\underline{5.45}$ cubic feet.

The volume of air to be supplied per cubic foot of fuel-vapor . . . . . $x\,a = 16.35$ cubic feet.

The total volume of the mixture, per cubic foot of fuel-vapor . . . . $x\,a + 1 = 17.35$ cubic feet.

The volume of the expanded normal charge after completed suction-stroke; using the value $\frac{V_a}{V_o} = 1.1$

$$\frac{V_a}{V_o}(x\,a + 1) = 1.1 \times 17.35 = 19.08 \text{ cubic feet.}$$

The heating-value per cubic foot of vapor $H = 1111$ B.T.U.

The heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{1111}{19.08} = 58 \text{ B.T.U.}$$

With a compression ratio of 7 to 1, that should be used in an alcohol-engine, there should, according to Table IV, be realized an efficiency approximately 0.31. In practice, this efficiency has been well exceeded under good average conditions. Counting, therefore, on an efficiency 0.31, the minimum required suction-displacement per minute, per I.H.P., will be

$$D_1 = \frac{42.42}{E\,f\,y \times \dfrac{H}{\frac{V_a}{V_o}(x\,a + 1)}} = \frac{42.42}{0.31 + 58}$$

$$D_1 = 2.36 \text{ cubic feet per minute.}$$

The required suction-displacement per minute per rated I.H.P., allowing 15 per cent overload capacity, is

$$D_2 = 1.15 \frac{42.42}{0.31 \times 58} = 2.71 \text{ cubic feet per minute.}$$

The required suction-displacement per minute per rated B.H.P., assuming the mechanical efficiency to be 0.85, is

$$D_4 = 1.35 \frac{42.42}{0.31 \times 58} = 3.19 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions, when operating on denatured alcohol-fuel, will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{3.19 \times 3{,}545} = \frac{l\,a\,N}{11{,}300}$$

l being the length of the stroke in inches, a the area of the piston in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will, according to equation 50*b*, be

$$\text{M.E.P.} = 98 \text{ pounds.}$$

CHAPTER VIII

FEATURES OF THE PRACTICAL GAS-ENGINE CYCLE

Ignition.—In order to explode a fuel-mixture it is required only that a small portion of the fuel be heated to the ignition-temperature. The heat of combustion of only a small portion of the explosive mixture will cause the inflammation of the whole charge, either through ordinary flame-propagation or possibly by means of an explosive wave which is assumed to cause practically instantaneous combustion of the whole charge.

The ignition-temperatures, which have been established by various experimenters, for hydrogen, carbon monoxide, marsh gas and ethylene are approximately:

For H	mixed with O	1100° Fahr.
For CO	mixed with O	1300° Fahr.
For CH_4	mixed with O	1200° Fahr.
For C_2H_4	mixed with O	1100° Fahr.

The diluting of a mixture does not generally change its ignition-point, excepting in the case of CO gas mixed with CO_2, in which case the ignition-point becomes materially higher. The ignition of the various gases in air occurs, thus, approximately at the same temperature as when mixed with oxygen.

The Timing of the Ignition.—The ignition of the charge should be timed to suit the fuel, the compression, and the speed of the engine. The surest means for finding the best point for ignition is the indicator, but it can generally be determined approximately by listening to the sound in the engine while the timing-device is varied.

In Fig. 37, I, II and III, are represented cards from an engine running on natural gas, taken at full load with the ignition timed differently in each case, and they show:

Card I a correct ignition.

Card II a too early ignition.

Card III too late ignition.

Card IV is taken at a lighter load and with an ignition entirely too late to suit the low compression of the charge.

Flame Propagation.—The rate of the flame propagation in a perfectly vaporized explosive mixture of normal composition is very high, and ordinarily no essential advantage would be derived

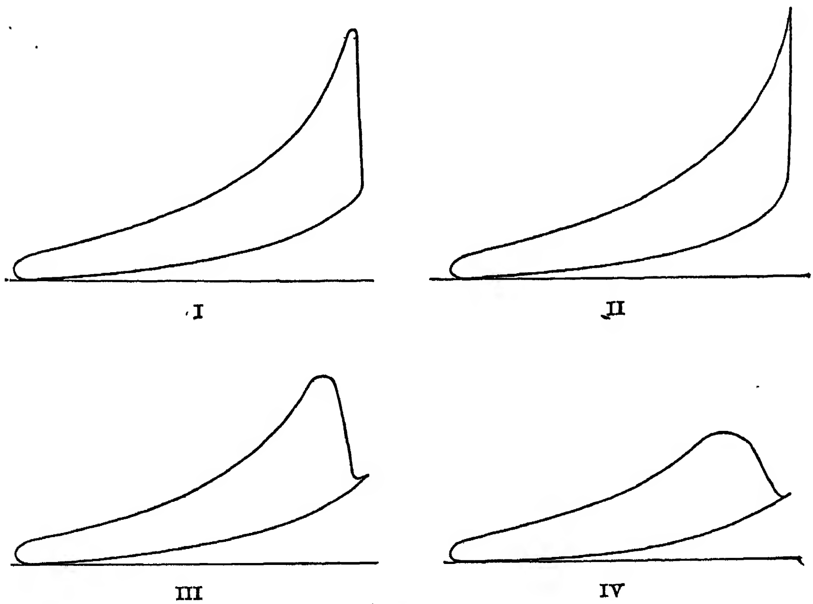


FIG. 37.

by an increased rate. In a diluted mixture, however, the rate is often not what would be desired.

Experiments by Mr. Grover,* at Yorkshire College, show that the replacing of the excess air in a diluted coal-gas mixture, in part, with burned gases (neutrals), may increase the rapidity of its combustion. Experimenters are not united, however, as to what influence the neutrals have on the rapidity of combustion of an explosive mixture, in general, though with respect to some fuels the detriment of replacing excess air for neutrals seems to be proven.

* "Modern Gas and Oil Engines," by Frederick Grover.

Mr. Grover's experiments establish the following additional conclusions with respect to a coal-gas mixture:

That the highest pressures are obtained when the volume of air is only slightly in excess of the amount required for complete combustion.

That when the volume of the products of combustion does not exceed 58 per cent of the total mixture then the mixture is still explosive, provided the volume of air is not less than 5½ times the volume of gas.

That higher pressures are recorded when the residual gases take the place of excess air.

The above conclusions, with regard to the experiments, amount practically to the same as saying that a perfect coal-gas-and-air mixture may be diluted with 140 per cent of its volume of burned gas and still be explosive; and the velocity of the flame-propagation of such a mixture is more rapid than if air were used for diluent.

The temperature of the charge when ignition takes place has also influence on the rate of the flame-propagation, so that the higher the temperature the more rapid the combustion becomes.

Experience also shows that mixtures in which the fuel is imperfectly vaporized often give unsatisfactory results due to slow combustion, and that a charge too weak to explode with a normal spark can be made to explode by using a heavy spark.

Explosion Experiments.—Clerk's experiments on explosive mixtures are of interest as showing the rapidity with which combustion occurs.

These experiments were made with Glasgow and Oldham city-gas with varying proportions of air. The ignition of the charge was effected without previous compression, at 60° F., and the results are shown in Table XV. The highest pressure attained was, as will be seen, 91 pounds per square inch.

Some experiments on mixtures of hydrogen and air are given in Table XV *a*.

For Koerting's experiments with coal-gas under a moderate pressure see Table XV *b*.

TABLE XV

Proportion by Volume. Gas.	Proportion by Volume. Air.	Maximum Observed Pressure. Pounds per sq. in.	Time to Reach Maximum Pressure. Seconds.
1	14	40	0.45
1	13	51.5	0.31
1	12	60	0.24
1	11	61	0.17
1	10	.....	.....
1	9	78	0.08
1	8	....	.....
1	7	87	0.06
1	6	90	0.04
1	5	91	0.055
1	4	80	0.16

TABLE XV *a*

Proportion by Volume. Hydrogen.	Proportion by Volume. Air.	Maximum Observed Pressure. Pounds per square inch.	Time to Reach Maximum Pressure. Seconds.
1	6	41	0.15
1	4	68	0.026
2	5	80	0.01

TABLE XV *b*

Proportion by Volume. Gas.	Proportion by Volume. Air.	Pressure Before Ignition. Pounds per sq. in. Absolute.	Time to Reach Maximum Pressure. Seconds.	Velocity of the Propagation of Pressure. Ft. per second.
1	7.5	15.0	0.032	23.0
		37.0	0.036	20.4
1	5.42	15.0	0.01	44.0
		37.0	0.0125	59.0

The propagation of the flame in the experiments by Koerting is essentially higher than in those of Clerk, which undoubtedly is due to differences in the general conditions under which the experiments were made. Comparing each set of results in Table XV *b* it will be seen that the flame-propagation is somewhat more rapid at the low pressures than at the higher ones. This may be according to what would be expected, as there is more weight of gas to be consumed in the latter case than in the former. The consumption of a unit weight of gas is, however, much more rapid at the higher pressure.

The following tables XVI and XVII are the results of experiments on explosive mixtures made at the Massachusetts Institute of Technology, and in Fig. 37 *a* are constructed, on a time basis, the pressure-lines obtained from every second of the mixtures of Table XVI; beginning with the 1 to 3 mixture.

TABLE XVI.

Results of Tests on Explosive Mixtures of Illuminating-Gas and Air.

Mixture (by parts Vol.).	Maximum Pressure. Pounds per sq. in.	Time of Explosion. Seconds.	First ⅛ Second.				⅛ Second after Maximum Pressure.				
			Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.	Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.	Final Pressure ÷ Proportion of Gas.
Gas-air											
1– 3	45	.49	0.32	11	44	26	1.80	43	172	40	160
1– 4	86	.08	1.77	59	295	61	1.88	62	310	46	230
1– 5	96	.05	1.86	62	372	52	1.93	64	384	44	264
1– 6	88	.05	1.80	60	420	54	1.93	64	448	46	322
1– 7	86	.06	1.97	66	528	58	1.93	64	512	46	384
1– 8	87	.06	1.71	57	513	53	1.83	61	549	46	414
1– 9	77	.08	1.60	53	530	57	1.86	62	620	46	460
1–10	71	.11	1.36	45	495	56	1.69	56	616	45	495
1–11	68	.14	1.21	40	480	60	1.66	55	660	43	516
1–12	39	.33	0.35	12	156	29	0.98	33	429	30	390
1–13	32	.42	0.18	6	84	16	0.79	26	364	24	336
1–14	9	.42	0.05	2	30	4	0.24	8	120	8	120

It will be noticed by the diagrams, Fig. 37 *a*, that, generally, the quicker the combustion occurs, and the higher the initial pressure, the lower the pressure-line becomes after one second's time, excepting in the case of the very rich mixture, curve 3. The end of the first one-fifth second from the time of the ignition is marked in the figure by a heavy line at the point $\frac{12}{60}$ of the total length of the diagrams, and at that line the final pressures

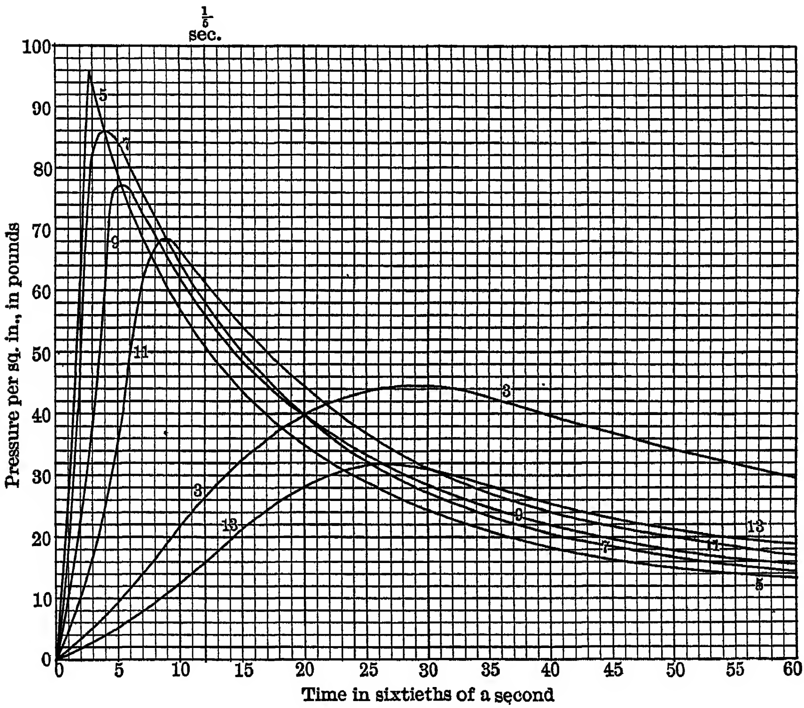


FIG. 37 *a*.

after the first one-fifth second are measured. The areas obtained after the first one-fifth second, and the mean pressures, columns 4 and 5 of Table XVI, refer also to the areas of the diagrams up to that line. The figures of the sixth and tenth columns of the tables represent, of course, the efficiencies at the combustion of each of the mixtures.

TABLE XVII.

Results of Tests on Explosive Mixtures of Gasoline (76°, Sp. gr. 0.680) and Air.

Proportion of Gas in Mixture.	Time of Explosion. Seconds.	First 0.2 Second.				Maximum Pressure.	0.2 Second after Max. Pressure.			
		Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.		Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.
0.0132	.167	.76	25.3	1925	52	52	1.28	42.7	3240	33
0.0141	.117	1.15	38.4	2720	49	62	1.42	47.3	3360	35
0.0151	.109	1.26	42.0	2770	48	64	1.45	48.6	2950	35
0.0164	.182	.81	27 0	1650	50	51	1.25	41.7	2540	32
0.0179	.109	1.27	42.3	2368	50	67	1.53	51.0	2855	36
0.0196	.091	1.44	48.0	2441	48	73	1.53	51.0	2600	36
0.0217	.082	1.43	47.7	2180	48	76	1.56	52.0	2391	37
0.0244	.060	1.62	54.0	2213	45	85	1.63	54.3	2225	36
0 0263	.058	1.61	53.7	2040	45	85	1.62	54.0	2052	36
0.0278	.058	1.62	54.0	1943	46	84	1.64	57.4	1970	38
0.0303	.066	1.49	49.7	1640	45	78	1.60	53.4	1760	37
0.0323	.067	1.55	51.7	1602	48	83	1.70	56.7	1760	38
0.0345	.100	1.34	44.7	1297	52	75	1.59	53.0	1536	38
0.0385	.117	1.10	36.7	955	52	62	1.42	47.3	1230	35
0.0417	.133	.98	32.7	761	52	55	1.40	46.7	1121	38
0.0476	.210	.39	13.0	273	35	35	1.02	34.0	714	32

Mixtures Highly Diluted with Combustion-Products.—It may appear from Mr. Grover's experiments with mixtures diluted with air and burned gases as if the neutrals in a weak mixture would tend to facilitate rather than retard the inflammation of the charge. However, the conditions under which these experiments were carried out were materially different from those existing during the combustion in the gas-engine cylinder, and experience, with some fuels at least, shows results differing in this respect. What effect, then, will the combustion-products actually have on the combustion in the gas-engine cylinder?

Assume a case of a throttling engine of fairly high compression,

which, due to faulty valve-setting, does not relieve itself of the burned gases to the best advantage. Such an engine we find, on a heavy load, apt to give evidence of pre-ignitions, which are due probably to the excess heat the burned gases transmit to the charge. Under a light load, however, the operation of the engine is liable to be troubled by back-firing into the mixing-chamber or inlet valve-casing, at times of the opening of the inlet value for admission of the new charge. This feature would be explained on the ground that an excessive amount of combustion-products in a weak mixture makes it slow-burning, to the extent of holding the fire all through the cycle until the inlet valve begins to open for a following suction-stroke.

After readjusting the valve-setting, by opening the exhaust valve early and keeping it open as long as practical, an appreciable change will have been accomplished. The cylinder will become scavenged from combustion-products as far as possible; hence the weak charge, on the same light load as before, will be diluted by more air instead of by an excess of combustion-products, and the back-firing into the mixing-chamber will have been cured. This tends to show, that of the two mixtures of similar heating-value the latter, containing less combustion-products, acts, in the working cylinder, as if it were the more inflammable.

Suppression of Heat at Combustion.—The specific heats of gases have been found to be approximately constant for temperatures as high as 500 degrees Fahr., and it has generally been assumed to be so for all temperatures. The heating-value admitted to the gas-engine cylinder with each pound of charge being known, the increase in temperature through combustion should, according to this assumption, theoretically be

$$t_f - t_i = \frac{Q}{c_v}.$$

However, judged by the corresponding increase in the pressure, this increase in temperature is never realized in practice, and a phenomenal suppression of from 30 to 40 per cent of the total heating-value of the fuel is often observed.

There has existed, and still exists perhaps, to some extent,

differences of opinion as to the cause of this discrepancy between the maximum temperature that should theoretically be obtained at the combustion in a gas-engine cylinder and the temperature actually obtained. One of the theories regarding its cause assumes that the metal of the combustion-chamber absorbs the heat apparently lost, later restoring part of it to the charge during its expansion; the effect being similar to that due to the so-called "wall-action" in a steam-engine cylinder.

The cause has, by another theory, been sought in the fact that some of the combustion-products become dissociated at very high temperatures, and that, therefore, the temperature obtained must always be below that at which dissociation takes place. According to a third theory, the suppression of heat is due to slow combustion, which would have for effect the lowering of the temperature through expansion before the full heating-value becomes evolved by combustion.

That the wall-action of the cylinder has some influence on the temperature obtained at the combustion there is no doubt, but, as a following approximate determination of the same will tend to show, it cannot be very considerable. It has, however, a decided tendency to absorb a certain amount of heat at the early part of the explosion-stroke, and to restore it, partly, later during the expansion. Clark's dissociation theory has lately been proven incorrect, on the ground that the temperature in the cylinder never attains that intensity at which either the carbonic acid gas or steam are dissociated. Finally, the after-burning theory, although it has full force with respect to certain slow-burning mixtures, does not seem to be true with respect to mixtures giving an average good combustion. Entropy-temperature diagrams constructed from indicator cards showing proper combustion-lines do not exhibit from the time of maximum pressure any additional heat-energy above what would readily be due to the action of the cylinder walls.

At present, when it appears certain that the specific heat of all gases formed at the combustion in the cylinder increases with the temperature, the so-called "heat suppression" would seem to have found a solution. And, with the assumption that the

values of the specific heat of gases at high temperatures which are at hand up to date are approximately correct, it can readily be ascertained that the temperatures and pressures obtained in the gas-engine are not materially lower than what can be expected theoretically.

According to determinations made by Mallard and Le Chatelier, the specific heats for the gases quoted below increase with the temperature in the following rate; the temperature being expressed in degrees Fahr.

TABLE XVIII

The specific heat at constant pressure.

$$\begin{aligned} CO_2 &= 0.185 + 0.000093\,t \\ H_2O &= 0.415 + 0.000202\,t \\ N &= 0.240 + 0.000024\,t \\ O &= 0.211 + 0.000021\,t \\ \text{Air} &= 0.233 + 0.000023\,t \end{aligned}$$

The specific heat at constant volume.

$$\begin{aligned} CO_2 &= 0.140 + 0.000093\,t \\ H_2O &= 0.306 + 0.000202\,t \\ N &= 0.173 + 0.000024\,t \\ O &= 0.150 + 0.000021\,t \\ \text{Air} &= 0.165 + 0.000023\,t \end{aligned}$$

Assume that in an engine there is used as fuel gasoline, the lower heating-value of which is 18,500 B.T.U., and that the compression ratio is $r = 4$.

Thus, $r^{n-1} = r^{0.35} = 1.62$.

The normal expanded charge after completed suction-stroke, including 15 per cent excess air, contains under these conditions, at about 80 degrees F., the heating-value (see page 127)

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = 70 \text{ B.T.U. per cubic foot.}$$

Hence, the heating-value per pound of normal charge will be, according to equation 44,

$$Q = \frac{H}{\frac{V_a}{V_o}(x\,a + 1)} V_a \frac{r-1}{r} = 70 \times 13.5 \times 3/4 = 708 \text{ B.T.U.}$$

The maximum temperature obtained at the combustion has been found by investigators to be, ordinarily, between 3,000° and 3,800° Fahrenheit—on an average thus 3,400° Fahrenheit.

The absolute temperature after compression is

$$T_b = r^{n-1}\,T_a = 1.62\,(460 + 80) = 875° \text{ F.}$$

or 415° above Fahrenheit zero.

The mean temperature during the combustion may, therefore, be called, approximately 1,900° F.

If this temperature, 1,900° F., be inserted in the preceding equations for the specific heat at constant volume, for CO_2, H_2O, N and O, we obtain the following:

The mean specific heat at constant volume for a range of temperatures between 415° and 3,400° F., for

	S_v
CO_2	= 0.317
H_2O	= 0.690
N	= 0.218
O	= 0.190

At the combustion of one pound of gasoline in 15 per cent excess air, there are formed the following products:

CO_2	3.0	pounds
H_2O	1.4	pounds
N	13.0	pounds
O	0.5	pounds
Total	17.9	

The average specific heat at constant volume, S_v, of all the products of combustion, therefore, for a range of temperatures between 415° and 3,400 F., is

Combustion Products, lbs.		S_v		
3.00	×	0.317	=	0.951
1.40	×	0.690	=	0.966
13.00	×	0.218	=	2.834
0.50	×	0.190	=	0.095
				4.846

$$S_v = \frac{4.846}{17.9} = 0.27$$

Inserting the preceding values for Q, S_v and $r^{n-1} T_a$ in equation 43, which may be written:

$$\frac{P_c}{P_b} = 1 + \frac{fQ}{S_v r^{n-1} T_a}, \text{ we obtain}$$

$$\frac{P_c}{P_b} = 1 + \frac{f}{0.27} \; \frac{708}{875} = 1 + 3.00 f.$$

Assuming that during the combustion there will be no loss of heat to the cylinder walls, thus $f = 1$, then

$$\frac{P_c}{P_b} = 4.00$$

In a non-scavenging engine this value, $\frac{P_c}{P_b} = 4$, is obtained very rarely, and for ordinary running a very good result would be $\frac{P_c}{P_b} = 3.75$, which corresponds to an initial pressure 304 pounds above the atmosphere.

Assuming $\frac{P_c}{P_b} = 3.75$ to be a normal value for good cards, then f becomes 0.917, showing a loss to the combustion-chamber walls of 8.3 per cent of the total heating-value.

The maximum temperature at the combustion will be obtained from equation $fQ = S_v (T_c - T_b)$, which for $f = 0.917$ becomes

$$0.917 \times 708 = 0.27 (T_c - 875).$$

Thus, $T_c = 3280°$ F.

This value is near enough to the assumed value $T_c = 3400°$ F., on which the computation for the mean S_v was based, to make it unnecessary to carry through any correction.

Heat-loss at the Combustion.—It may be possible to get an approximate idea about the amount of heat that reasonably can be expected to dissipate into the metal of the combustion-chamber, during the short time the combustion is in progress, through comparison with the total heat that becomes dissipated into the water-jacket during the time of the whole cycle.

Assume a 24 × 36 single-acting engine of a piston speed of 600 feet per minute. Thus, 100 revolutions per minute, or 1⅔ revolution per second.

The indicated power of this engine is

$$\frac{A\,S}{4D_1} = \frac{314 \times 600}{4 \times 2.55} = 180 \text{ I.H.P.}$$

Hence the heating-value transferred into indicated horse-power is:

$$\begin{array}{lll} & \text{per minute } 180 \times 42.42 = & 7636 \text{ B.T.U.,} \\ & \text{per second} \qquad\qquad = & 127 \text{ B.T.U.,} \\ \text{or} & \text{per cycle} \qquad\qquad\;\; = & 152.72 \text{ B.T.U.} \end{array}$$

From tests it is known that in an engine of this class there is dissipated into the water-jacket, generally, heat amounting to somewhat more than that transformed into indicated power. Assume that 10 per cent more heat is dissipated than that obtained as indicated power. Thus, a total of 168 B.T.U. per cycle.

Add to this heat-energy again 10 per cent of the heating-value accounted for by the indicator card, or 15 B.T.U., to allow for the heat which is returned to the charge during the later part of the expansion-stroke. This percentage is ample as, actually, entropy-temperature diagrams generally show not much over one-half of this amount.* Add also the heat which the incoming charge absorbs, from the cylinder, which, according to page 113, is about 24,300 B.T.U. per hour, or, 8 B.T.U. per cycle.

168 + 15 + 8 (= 191) B.T.U. is, accordingly, abstracted from the working gases, during the latter part of the compression-stroke, during the explosion and during the beginning of the expansion-stroke. During the latter part of the expansion-stroke

* Compare "Entropy Analysis of the Otto Cycle." S. A. Reeve, Transact. Am. Soc. Mech. Engs., Vol. XXIV.

and during the exhaust-stroke the gases are of such an essentially lower temperature than at the early part of the expansion that no heat can then pass from them into the metal—the opposite might rather be the case, at least during part of the period.

Assume, therefore, that all the heat is abstracted during one-half of one revolution, or during $\frac{1}{2} \times \frac{3}{5} = \frac{3}{10}$ second. Thus, 191 B.T.U. is abstracted in $\frac{3}{10}$ of a second, or at the mean rate of 637 heat-units per second.

The mean temperature of the gases during the combustion is, of course, practically the same as their mean temperature during the expansion, but, the metal of the combustion-chamber being at the former period comparatively cold, it is to be assumed that the rate at which heat is abstracted then is much greater than during the time of expansion. A fair assumption would, probably, be that the rate of which heat is abstracted during the combustion is twice the mean rate at which it is abstracted during the entire period of heat-loss.

At that rate there would be abstracted during the combustion 1,274 heat-units per second, and as the average time for a proper explosion is, say, 0.04 of a second, the total amount of heat dissipated during the explosion becomes 50 heat-units. Assuming the thermal efficiency to be 25 per cent, then the heat dissipated into the metal of the combustion-chamber becomes $\frac{50 \times 0.25}{152.72} =$ 8 per cent of the total heating-value.

The above results are, of course, founded on several assumptions that cannot very well be proven to be exactly correct, but if the assumptions are reasonable the results will tend to show that the heat-suppression observed, in connection with any good combustion in the cylinder, can very well be explained, since the specific heat increases materially at high temperatures, and since the metal of the combustion-chamber actually absorbs, during the combustion, from 5 to 10 per cent of the total heating-value liberated.

The Effect on the Expansion-Line of the Diluting of the Charge.—In Fig. 38 are drawn three curves, the first an isothermal curve, the second and third adiabatics for $n = 1.25$ and for

$n = 1.3$. It is evident that the smaller the ratio $\frac{c_p}{c_v} = n$ is the nearer to the isothermal line the expansion line will fall, and the greater the M.E.P. and area of the indicator card becomes.

In a card from a diluted mixture, the mean effective pressure is often higher, relatively to the initial pressure, than in a card from a normal mixture, and this may be thought to be due to the difference in the specific heat of the combustion-products from the diluted mixture and from the normal mixture. If it be assumed, however, that the specific heats of the elementary gases increase with the temperature in the rate Table XVIII, page 173, shows then, this cannot be the case.

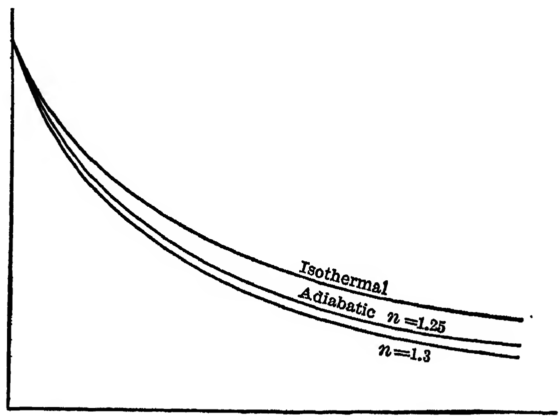


FIG. 38.

Computing, similarly as on page 175, the specific heats of the products of combustion from gasoline, including 15 per cent excess air, at the mean temperature 1,900° F., we obtain the following:

	Combustion Products, lbs.	c_p		Combustion Products, lbs.	c_v	
CO_2	3.0	× 0.362	= 1.086	3.0	× 0.317	= 0.951
H_2O	1.4	× 0.799	= 1.118	1.4	× 0.690	= 0.966
N	13.0	× 0.286	= 3.718	13.0	× 0.218	= 2.834
O	0.5	× 0.251	= 0.125	0.5	× 0.190	= 0.095
	17.9		6.047	17.9		4.846

$$c_p = \frac{6.047}{17.9} = 0.338 \qquad c_v = \frac{4.846}{17.9} = 0.27$$

$$\frac{c_p}{c_v} = 1.25 = n.$$

On the other hand, assume that the mixture be diluted with the same quantity of air that is actually required for its combustion; it accordingly being charged with about 15 pounds excess air. The mean temperature at the combustion will then be considerably less than before, approximately only 1,500° F.

The computation for the specific heat of the products of combustion of gasoline-fuel diluted with 100 per cent excess air, at the mean temperature 1,500° F., will be:

	Combustion Products, lbs.	c_p		Combustion Products, lbs.	c_v	
CO_2	3.0	× 0.324	= 0.972	3.0	× 0.279	= 0.837
H_2O	1.4	× 0.718	= 1.005	1.4	× 0.609	= 0.853
N	11.6	× 0.276	= 3.202	11.6	× 0.209	= 2.424
Air	15.0	× 0.267	= 4.005	15.0	× 0.199	= 2.985
	31.0		9.184	31.0		7.099

$$c_p = \frac{9.184}{31} = 0.296 \qquad c_v = \frac{7.099}{31} = 0.229$$

$$\frac{c_p}{c_v} = 1.3 = n.$$

As the lowest of the curves, Fig. 38, represents the expansion-line for a diluted charge, it is evident that the change in the specific heat of the combustion-products, due to the diluting of the charge, does not have for effect the raising of the expansion-line; rather the contrary. That the expansion-line from a diluted charge often is higher relatively to the initial pressure than the expansion line from a normal charge depends, it must be concluded, on some other cause apart from its dilution.

The Relation between Initial- and Mean Effective Pressure.—The value of the coefficient y is, according to equation 53, page 40,

$$y = \frac{m.e.p.\ (n-1)\ (r-1)}{E\ (p_c - p_b)}$$

In order to give this formula a practical application, we solve the value of y for each of the following six cards from an engine, under test, working on kerosene and gasoline fuels. The engine, an Otto hit-or-miss, $6\frac{3}{4} \times 15\frac{1}{2}$, 15 brake horse-power engine, was arranged with a special vaporizer for using kerosene.

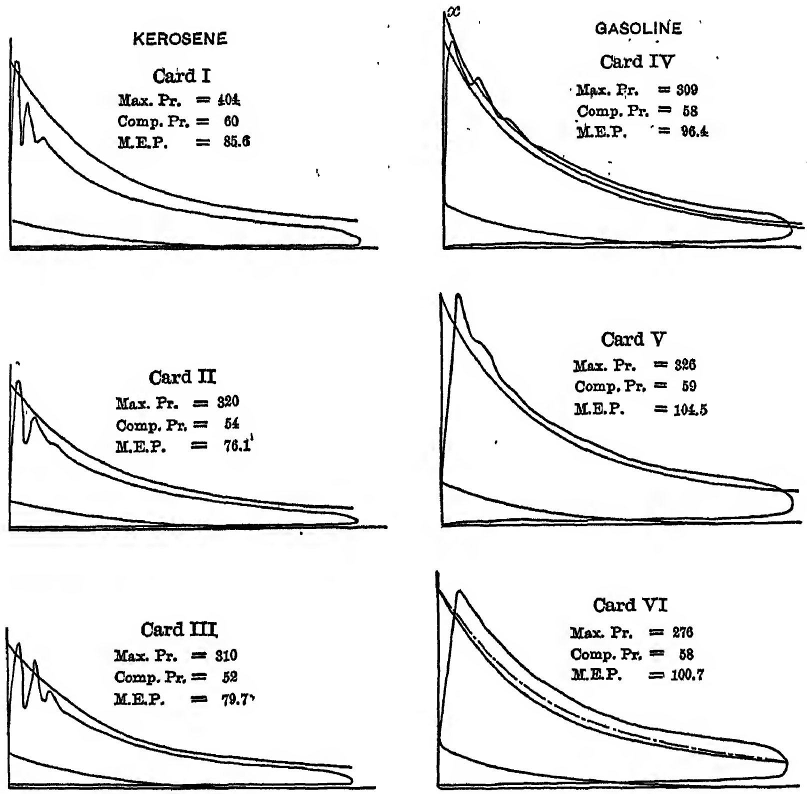


FIG. 39.—CARDS FROM ENGINE-TEST ON GASOLINE AND KEROSENE.

The maximum-, the mean effective-, and the compression-pressures are noted on each card.

The actual compression ratio of the engine is 3.9 to 1, which, compared with the compression pressures obtained, shows that the charge, at the beginning of the compression stroke, was of a pressure somewhat less than that of the atmosphere. We obtain the factor $r - 1 = 2.9$.

The value of the coefficient E, on the basis of a compression ratio 3.9 to 1, is $E = 0.372$ and $n - 1 = 0.35$, approximately.

Hence, the factor $\frac{(n-1)\ (r-1)}{E} = 2.72.$

Inserting the values of $m.e.p.$ and $p_c - p_b$ in the equation $y = 2.72 \frac{m.e.p.}{p_c - p_b}$, and solving for y we obtain the following:

	Card.	M. E. P.	p_c.	p_b.	$p_c - p_b$.	y.
Kerosene...	I	85.6	404	60	344	0.7
	II	76.1	320	54	266	0.8
	III	79.7	310	52	258	0.83
Gasoline ...	IV	96.4	309	58	251	1.05
	V	104.5	326	59	267	1.07
	VI	100.7	276	58	218	1.26

If the figures for the cards II and III of the first group be compared we find that, although the initial pressure of card III is the lowest, yet its mean effective pressure is the highest. This we find more in evidence by comparing the figures for cards IV and VI of the second group. VI having, by far, the lowest initial pressure, yet, its mean effective pressure is the highest.

The mean effective pressure of the actual indicator cards we find, thus, essentially independent of the maximum pressure, and the actual area of the card is, as the coefficient y shows, from 30 per cent less to 26 per cent larger than the area of the theoretical card. The variation in the value of the coefficient y we find to be, to some extent, due to the fact that the maximum initial pressure is more or less pushed back away from the head-end of the card.

On each card of Fig. 39 there is drawn an expansion line following the equation $p = \left(\frac{v_1}{v}\right)^{1.35} p_1$, by which the actual and the theoretical expansion lines, as well as the areas of the actual and the theoretical cards, may be compared.

The nature of the fuel, the temperature of the charge, the

proportioning of the mixture, as well as the timing of the ignition, all have influence on the slope of the combustion line, as well as on the appearance of the card at the high point of maximum pressure. Thus we find that the kerosene cards, which were taken with the fuel- and air-mixture arriving to the engine heated to a temperature not less than 600° F., have a much quicker combustion-line, and are sharper at the high point than the gasoline cards, which were taken with the fuel arriving cold and probably less thoroughly vaporized. The expansion lines of the kerosene cards drop, however, quickly below the theoretical expansion line, which evidently is due to the quicker combustion and consequently higher momentary combustion-temperature, whereas the expansion lines of the gasoline cards all show a tendency to keep above this line.

The correctness of the information that the coefficient y will give is not absolute, but it will be close enough for a comparison between cards from different fuels, and particularly so if the cards are all taken from the same engine. It will be apparent, that the coefficient expresses simply the ratio between the mean effective pressure obtained from the actual card and the theoretical mean effective pressure of the air-card. In order, then, that its value shall be reasonably correct it will be required that we shall use in the formula the correct expansion ratio (this ratio is assumed to be the same as the compression ratio) and therefore the true clearance space in the engine must be known.

To illustrate the influence the employment of an incorrect expansion ratio would have on the result given by formula 53, there are in card VI, Fig. 39, drawn two curves, the lower being the correct one for an expansion ratio 3.9 to 1 and the upper for an expansion ratio of 3.5 to 1. The upper curve, enclosing a materially larger air-card area, will give a materially smaller value of the coefficient y. The difference will, however, be of minor importance when the object is simply to compare the different cards on the same basis.

For such cards as shown in Fig. 40, taken on a light load and with the charge throttled, or diluted, considerably, the coefficient y may become very large, but it has in such cases no significance

as the card, then, does not even approximately follow the theory laid down for the cycle. In others, it may be considerably less than the unit, giving evidence of a cold cylinder or leaking valves.

In card IV, through the point x, which, according to the theory, should have been the starting-point for the actual expansion curve, there is drawn a theoretical expansion line ($n = 1.35$). The area below this line is to some extent smaller than the area below the actual expansion curve, and the work represented by the area between the theoretical and actual expansion lines represents the heat added after the beginning of the expansion. This heat is evidently obtained from the heated metal of the

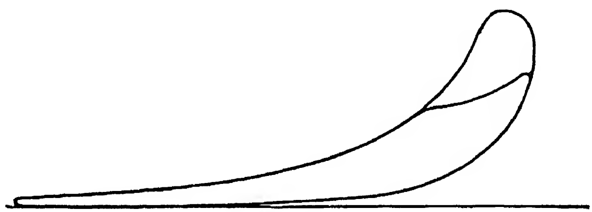


FIG. 40.

cylinder; and the fact that the actual expansion line often rises above the theoretical line toward the end of the stroke, after the cylinder has become well heated, may to some extent help to explain the phenomenon that heat appears to be suppressed at the beginning of the cycle to reappear, later, during the expansion.

At the tests of the kerosene and gasoline fuels, during which the cards, Fig. 39, were taken, the average fuel-consumption, and the efficiency of the heat transformation, were as follows:

	Kerosene.	Gasoline.
Fuel consumption per hour per I.H.P. lbs.	0.675	0.709
Fuel consumption per hour per B.H.P. lbs.	0.873	0.894
Average efficiency for all tests	20.21	17.9
Maximum efficiency at one test	24.89	20.7

The circumstance will be noticed that, although the coefficient y of the gasoline cards, and the cards themselves, appear much more favorable with respect to an efficient utilization of the heat, still, the kerosene gave actually a materially higher efficiency,

which fact tends to show that the individual indicator card, though it tells how effectively the heat was utilized after having been evolved in the combustion-chamber of the cylinder, cannot tell, excepting by comparison with some standard, how effectively the heating-value of the fuel was realized at the combustion.

Explosive Waves.—When indicator cards are taken from gas-

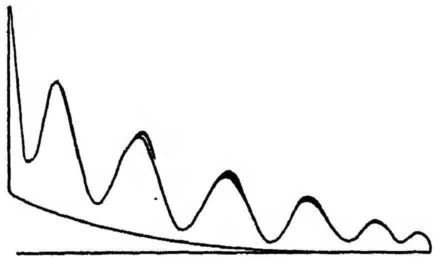


FIG. 41.

engines, there occasionally will appear cards, the expansion line of which shows irregularities that are hard to explain. Sometimes the expansion line, instead of being a smooth curve, becomes a wave-line consisting of a number of undulations, as Fig. 41. Berthelot, the French scientist, explained the phenomenon as

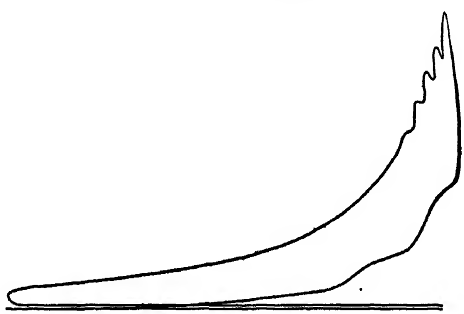


FIG. 42.

being due to, what he called, explosive waves in the cylinder. Investigations have shown that, if the charge becomes agitated during the compression, then the combustion is likely to be followed by violent waves, and experimenters have purposely produced agitation of the charge, by special means, in order to obtain such waves and to verify the cause of them.

In practice the agitation is assumed to originate from undulations in the charge set up through the action of the compression. Compression waves are occasionally observed on the indicator card, as in Fig. 42, and they are sometimes accompanied by more or less violent undulations of the expansion line, as at the

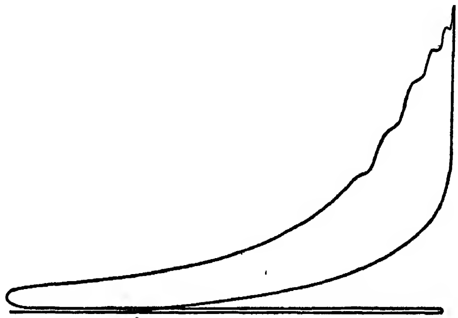


FIG. 43.

top of card, Fig. 42. It cannot be presumed that the waves indicated on the diagram represent, in every case, truly the fluctuations of pressure obtaining in the cylinder. That cannot be, because the inertia of the indicator pencil-arm and mechanism

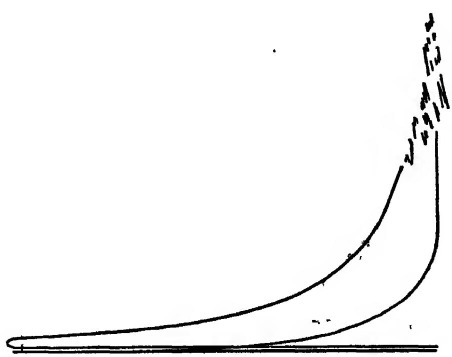


FIG. 44.

must exert a considerable influence on the wave-line indicated, due to the extraordinary rapid motion given to it. Wave-lines, such as shown in Fig. 43, may be entirely due to the inertia of the indicator pencil-arm and mechanism, just as corresponding waves are obtained on the steam-engine card.

There are often vibrations produced on the indicator-card that can readily be obliterated, by effecting an alteration in the connection between the indicator and the inside of the cylinder. Between the combustion-chamber and the tap for the indicator there is often a chamber for the accommodation of the valve that closes the indicator opening to the cylinder, when the instrument is removed. In this chamber there is likely to occur secondary explosions that distort the true expansion line due to the actual pressure in the cylinder, and, to prevent this, a pipe should be fitted, leading from the opening in the cylinder direct to the indicator.

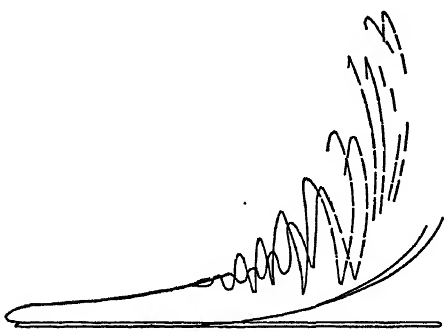


FIG. 45.

Sometimes, when the explosions are powerful, due to a rich mixture and early ignition, the top of the card will consist of a series of vibrations, which are evidently due to the springing of the instrument, and may be aggravated by the springing and vibrating of the metal of the engine forming the combustion-chamber. A card of this nature is shown in Fig. 44.

Occasionally the vibrations of the pencil-arm may be extremely violent, at the beginning of the stroke, and disappear during the expansion, resulting in a card as Fig. 45. This card was taken during two explosions and represents evidently a combination of vibrations due to pre-ignition and explosion waves.

This subject of explosion waves is not, at present, very fully understood, and engineers are at work trying to solve the mystery surrounding the matter. It is, however, apparent that the variety of explosion waves represented by the indicator is not due to a single cause only.

CHAPTER IX

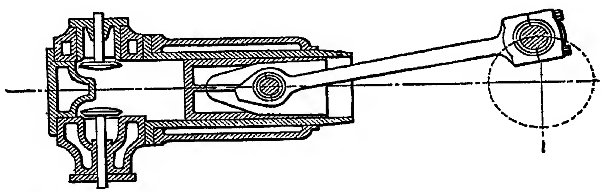
THE FLY-WHEEL

Four-Cycle Engine-Types.—The four-stroke cycle, or four-cycle, engine is built single-acting or double-acting, and is often arranged with two or more cylinders working on a common shaft.

With respect to cylinder arrangement, the following nine types are common for medium-sized and large stationary engines. Types II, III and VI are, however, less frequently used than the others.

TYPE I

SINGLE-ACTING ONE-CYLINDER ENGINE



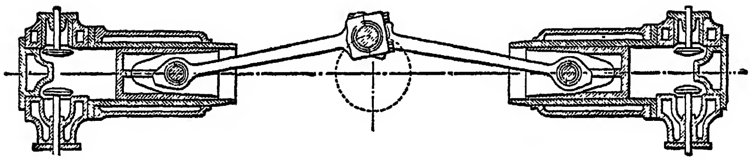
In this engine-type there occurs one pressure-stroke every fourth stroke of the piston. The cycle commences with the suction-stroke and the piston moving away from the valve-end of the cylinder. The next stroke becomes the compression-stroke, when the piston moves toward the valve-end. The third stroke is the expansion- or pressure-stroke, the piston moving away from the valve-end, and the fourth stroke is the exhaust- or discharge-stroke, when the piston again moves toward the valve-end of the cylinder. Each stroke corresponds to a course of 180 degrees by the swing of the crank; the whole cycle, therefore, occupying 4 times 180 degrees or 720 degrees.

The cycle is conveniently represented in a scheme as the following:

$$
\text{Valve End.}\left\{
\begin{array}{ccc}
 & \text{Suction} & \\
0^\circ & \xrightarrow{\qquad\qquad} & 180^\circ \\
 & \text{Compression} & \\
360^\circ & \xleftarrow{\qquad\qquad} & \\
 & \text{Expansion} & \\
 & \xrightarrow{\qquad\qquad} & 540^\circ \\
 & \text{Exhaust} & \\
720^\circ & \xleftarrow{\qquad\qquad} &
\end{array}\right.
$$

TYPE II

Single-Acting Two-Cylinder Opposed Engine



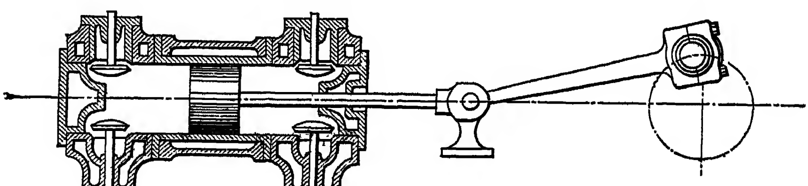
The scheme for the cycle will be:

Cylinder No. 1.	Cylinder No. 2.
Suspect: Suction → 180°	Compression → 180°
Compression ← 360°	Expansion ← 360°
Expansion → 540°	Exhaust → 540°
Exhaust ← 720°	Suction ← 720°
Valve End. (180°–540°)	Valve End. (0°–540°)

The expansion-strokes occur every second time 180 degrees and every second time 540 degrees apart.

TYPE III

Double-Acting One-Cylinder Engine



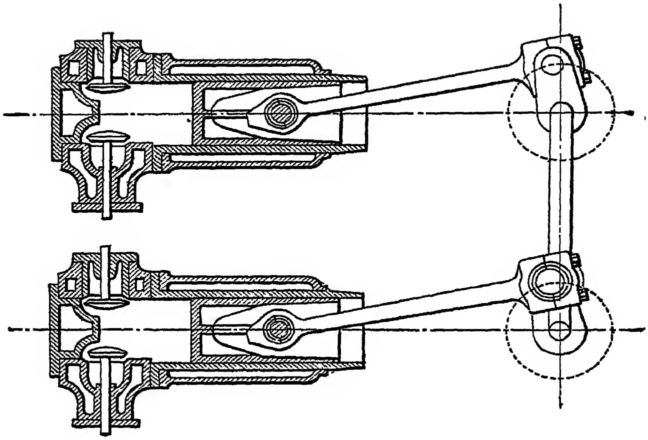
The scheme for the cycle will be:

Head End.	Crank End.
Suction → 180°	Compression → 180°
Compression ← 360°	Expansion ← 360°
Expansion → 540°	Exhaust → 540°
Exhaust ← 720°	Suction ← 720°
Valve End. (180°–540°)	Valve End. (0°–540°)

The expansion-strokes occur every second time 180 degrees apart and every second time 540 degrees apart, the same as in Type II.

TYPE IV

SINGLE-ACTING TWO-CYLINDER TWIN ENGINE



The scheme for the cycles will be:

Right-hand Cylinder.

Valve End.

Suction → 180°
360° ← Compression
Expansion → 540°
720° ← Exhaust

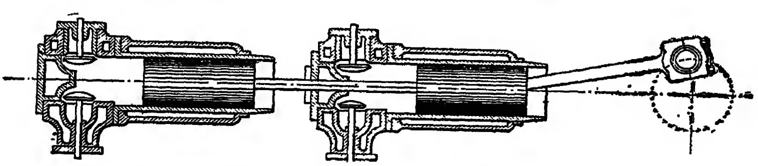
Left-hand Cylinder.

Valve End.

Expansion → 180°
360° ← Exhaust
Suction → 540°
720° ← Compression

TYPE V

SINGLE-ACTING TWO-CYLINDER TANDEM ENGINE



The scheme for the cycles will be:

Rear Cylinder.

Valve End.

Suction → 180°
360° ← Compression
Expansion → 540°
720° ← Exhaust

Front Cylinder.

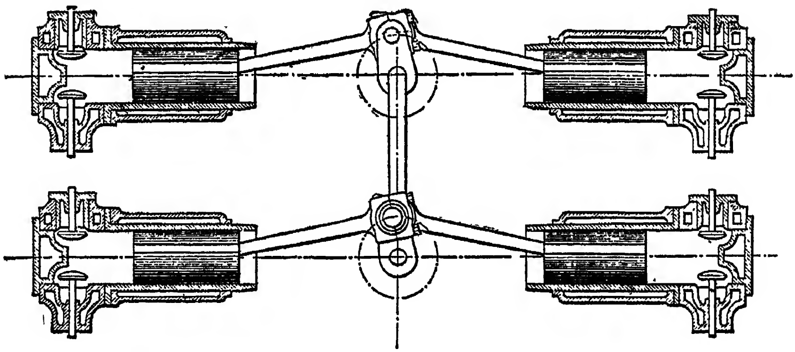
Valve End.

Expansion → 180°
360° ← Exhaust
Suction → 540°
720° ← Compression

The expansion-strokes of the two preceding engine-types occur uniformly 360 degrees apart.

TYPE VI

TWIN SINGLE-ACTING FOUR-CYLINDER OPPOSED ENGINE



The scheme for the cycles will be:

Left-hand Engine.

	First Cylinder.			Second Cylinder.	
Valve End.	Suspension → 180°			Exhaust → 180°	Valve End.
	360° ← Compression			360° ← Suction	
	Expansion → 540°			Compression → 540°	
	720° ← Exhaust			720° ← Expansion	

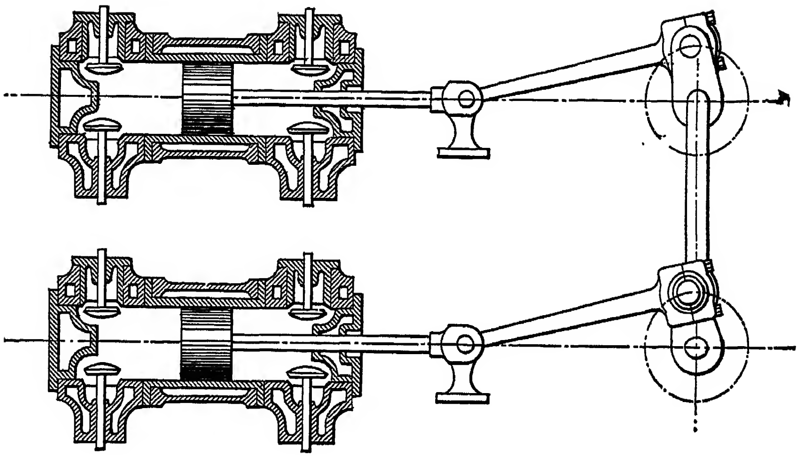
Right-Hand Engine.

	Third Cylinder.			Fourth Cylinder.	
Valve End.	Expansion → 180°			Compression → 180°	Valve End.
	360° ← Exhaust			360° ← Expansion	
	Suction → 540°			Exhaust → 540°	
	720° ← Compression			720° ← Suction	

The expansion-strokes occur every 180 degrees apart.

TYPE VII

Double-Acting Two-Cylinder Twin Engine



The scheme for the cycles will be:

Right-hand Cylinder.

Head End.		Crank End.	
Suction		Exhaust	
⟶	180°	⟶	180°
Compression		Suction	
360° ⟵		360° ⟵	
Expansion		Compression	
⟶	540°	⟶	540°
Exhaust		Expansion	
720° ⟵		720° ⟵	

Valve End. (180°–540°, Head End) Valve End. (180°–540°, Crank End)

Left-hand Cylinder.

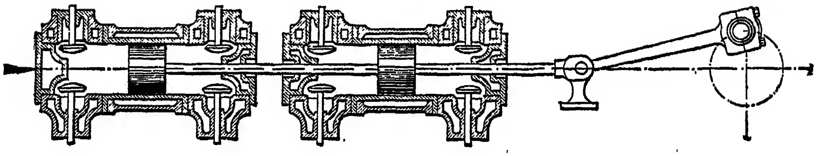
Head End.		Crank End.	
Expansion		Compression	
⟶	180°	⟶	180°
Exhaust		Expansion	
360° ⟵		360° ⟵	
Suction		Exhaust	
⟶	540°	⟶	540°
Compression		Suction	
720° ⟵		720° ⟵	

Valve End. Valve End.

The expansion-strokes occur 180 degrees apart the same as Type VI.

TYPE VIII

Double-Acting Two-Cylinder Tandem



The scheme for the cycles will be:

Front Cylinder.

	Head End.		Crank End.	
	Suction	→ 180°	Exhaust	→ 180°
Valve End.	Compression	← 360°	Suction	← 360°	Valve End.
	Expansion	→ 540°	Compression	→ 540°
	Exhaust	← 720°	Expansion	← 720°

Rear Cylinder.

	Head End.		Crank End.	
	Expansion	→ 180°	Compression	→ 180°
Valve End.	Exhaust	← 360°	Expansion	← 360°	Valve End.
	Suction	→ 540°	Exhaust	→ 540°
	Compression	← 720°	Suction	← 720°

The expansion-strokes occur every 180 degrees apart.

TYPE IX

Twin Double-Acting Tandem Four-Cylinder Engine

When the cranks are set 90 degrees apart, which is the most favorable for smooth running, there are obtained two sets of cycles according to the preceding scheme, and they are offset 90 degrees toward each other, giving one expansion-stroke every 90 degrees apart.

Fly-Wheel Theory.—The Tangential Crank-Effort.—The tangential crank-effort, or the force representing the driving effort at the crank-pin in a tangential direction to its orbit, varies materially for different positions of the crank. It is always zero at the beginning and at the end of each stroke, and between these points it may be in a direction so as either to accelerate or retard the forward motion of the wheel.

The variations in the tangential crank-effort for successive positions of the crank-pin may be represented, graphically, by a crank-effort curve plotted from the pressures known to exist in the cylinder for successive positions of the crank.

The Crank-Pin Pressure.—Fig. 46 is a normal indicator

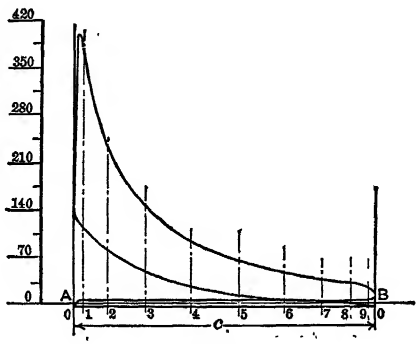


FIG. 46.

diagram representing the four-stroke cycle, on which are drawn nine ordinates perpendicularly to the atmospheric line *A B*. The distance between the atmospheric line and any of the four lines of the cycle, the suction-line, compression-line, expansion-line and exhaust-line, measured on any of the ordinates, represents the pressure on the piston at a corresponding point of the stroke. This pressure is, by means of the connecting-rod, transmitted to the crank-pin, but it is materially modified by a force evolved at the acceleration or retardation of the reciprocating parts. The latter force tends to relieve the pressure on the crank-pin at the beginning of each stroke, by subtracting from the effective pressure on the piston, and at the end of each stroke it tends to increase

the crank-pin pressure, by adding to the effective pressure on the piston.

The Accelerating Force.—The force required to accelerate (or to retard) the reciprocating parts at the head end of the piston-stroke is

$$P_1 = \pm \frac{12\,G}{g} \frac{V^2}{r} \left(1 + \frac{r}{l}\right),^* \quad . \quad . \quad . \quad (100h)$$

and the force required to accelerate (or to retard) the reciprocating parts at the crank end of the piston-stroke is

$$P_2 = \pm \frac{12\,G}{g} \frac{V^2}{r} \left(1 - \frac{r}{l}\right). \quad . \quad . \quad . \quad . \quad (100c)$$

At the point X, Fig. 47, where the velocity of the crosshead changes from an accelerating to a retarding one, the force is

$$P_x = \pm\, O. \quad . \quad . \quad . \quad . \quad . \quad . \quad (100x)$$

The notations in these equations are:

P_1, P_2, P_x the accelerating or retarding force in pounds;
G the total weight of the reciprocating masses;
V the velocity of the crank-pin, in feet per second, $= \dfrac{\pi r N}{360}$;
N the number of turns of the wheel, per minute;
g the acceleration due to gravity $= 32.16$;
r the crank-radius, in inches;
l the length of the connecting-rod, in inches.

The numerical factor of the equations for the force due to the acceleration of the reciprocating parts, for the following three ratios of r to l, becomes,

$\frac{r}{l}$	$\frac{1}{5}$	$\frac{1}{5.5}$	$\frac{1}{6}$
$\frac{12}{g}\left(1 - \frac{r}{l}\right) =$	0.448	0.441	0.435
$\frac{12}{g}\left(1 + \frac{r}{l}\right) =$	0.298	0.304	0.310

* For the derivation of this expression see page 472 of the Appendix.

The ratio $\frac{r}{l}$ being seldom more than $\frac{1}{5}$ nor less than $\frac{1}{6}$, we may say, with a fair approximation for any practical case, that

$$\frac{12}{g}\left(1+\frac{r}{l}\right) = 0.441,$$

and

$$\frac{12}{g}\left(1-\frac{r}{l}\right) = 0.304.$$

As, in the indicator diagram, pressures are represented by pounds per square inch of the piston area, it will be convenient to express the accelerating forces also in pounds per unit area of the piston. This may be done by dividing each side of equations 100*h* and 100*c* by the area of the piston, F, whereby is obtained:

The force, per square inch piston area, required for the acceleration (or retardation) of the reciprocating parts at the head end of the stroke.

$$\frac{P_1}{F} = \pm\, 0.441\, \frac{G}{F}\,\frac{V^2}{r}, \quad . \quad . \quad . \quad (100h\text{-}a)$$

and the force, per square inch piston area, required for the acceleration (or retardation) of the reciprocating parts at the crank end of the stroke.

$$\frac{P_2}{F} = \pm\, 0.304\, \frac{G}{F}\,\frac{V^2}{r}. \quad . \quad . \quad . \quad (100c\text{-}a)$$

The force, per square inch area at the point X

$$\frac{P_x}{F} = \pm\, O, \quad . \quad . \quad . \quad . \quad . \quad (100x\text{-}a)$$

If in these equations be inserted the value of $\frac{V^2}{r}$ expressed by the number of revolutions we obtain

$$\frac{P_1}{F} = \pm\, 0.000034\, \frac{G}{F}\, N^2\, r, \quad . \quad . \quad . \quad (101h)$$

and

$$\frac{P_2}{F} = \pm\, 0.000023\, \frac{G}{F}\, N^2\, r. \quad . \quad . \quad . \quad (101c)$$

These equations, it should be noted, are correct only when $l = 5\frac{1}{2}\, r$, but in any practical case they are sufficiently close approximations.

Tables of Weights of the Reciprocating Parts.—The weight, G, of the reciprocating parts for an engine not being definitely known, generally, at the time the crank-effort diagram is due to be laid out, it becomes convenient to give this factor an approximate value, in accordance with previous practice with engines of types similar to that of the one in hand.

The following tables give the usual weight of the reciprocating parts, per square inch piston area, for various classes of engines and for different cylinder sizes. From them may be selected a value for G suitable for most any case.

In the weight of the reciprocating parts there should properly be included ½ of the weight of the connecting-rod; ½ of its weight being counted as revolving. The weight, $\frac{G}{F}$, of the reciprocating parts per square inch of the piston quoted in the tables includes a proportionate part of the weight of the connecting-rod.

TABLE XIX.

For Automobile Engines

Cylinder diameter, inches:	3½	4	4½
Length of stroke, inches:	4	5	6
$\frac{G}{F}$ per each cylinder:	0.8	0.9	1.0.

TABLE XX.

For One-Cylinder Single-Acting Four-Cycle Engines

Diameter of cylinder.....	less than 6″	6″ to 12″	12″ to 18″	18″ to 24″
$\frac{G}{F}$	2 to 2.5	2.5 to 4	4 to 5	4.75 to 5.5

Large Engines with Water-cooled Pistons.

For double-acting, single-cylinder four-cycle engines: use for estimate 9 pounds per square inch piston area.

For double-acting, single-cylinder two-cycle engines: use for estimate 9 pounds per square inch piston area.

For double-acting, two-cylinder, tandem engines: use for estimate 17 pounds per square inch piston area.

The Acceleration Curve for the Reciprocating Parts.—In Fig. 47 are represented the paths for the piston-pin and crank-pin, with the crank and connecting-rod shown in the positions they occupy when the connecting-rod is tangential to the path of the crank-pin. The ratio $\frac{r}{l}$ may be assumed to be 5.5. At point A, representing the beginning of the outward stroke of the piston, we offset, on the negative side of the base-line, $A\ B$, and to the same scale as that of the indicator card, Fig. 46, $A\ C = \frac{P_1}{F} =$ the accelerating force per square inch of piston area, and at the point B, representing the end of the stroke, we offset, on the positive side of the base-line, $B\ D = \frac{P_2}{F}$. At X, at which point the connecting-rod stands tangential to the crank-pin circle, the velocity changes from an accelerating to a retarding one, and the accelerating force is in that point $\frac{P_x}{F} = O$.

The distance x of point X from the perpendicular M M, representing the middle of the piston travel, changes slightly for different ratios $\frac{r}{l}$. It becomes

for	$\frac{r}{l}$	=	$\frac{1}{6}$	$\frac{1}{5.5}$	$\frac{1}{5}$
	x	=	$0.07\,r$	$0.08\,r$	$0.09\,r$,

but as a mean it is, $x = 8$ per cent of r, and for convenience this value may be used in all cases.

After locating the point X, on the base-line $A\ B$, the proper distance from M M, we draw a smooth curve through the points C X D, and obtain then the curve of force due to the acceleration of the reciprocating parts. The accelerating force for the return stroke will be negative at B and positive at A, and the curve of force H X K due to the acceleration of reciprocating parts for the return stroke will cut the base-line $A\ B$ at X.

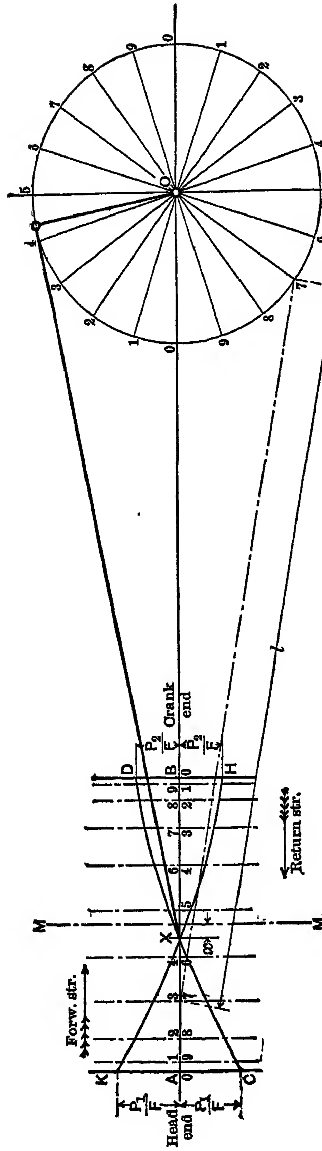

Forw. str.
Return str.
Head end
Crank end
l

FIG. 47.

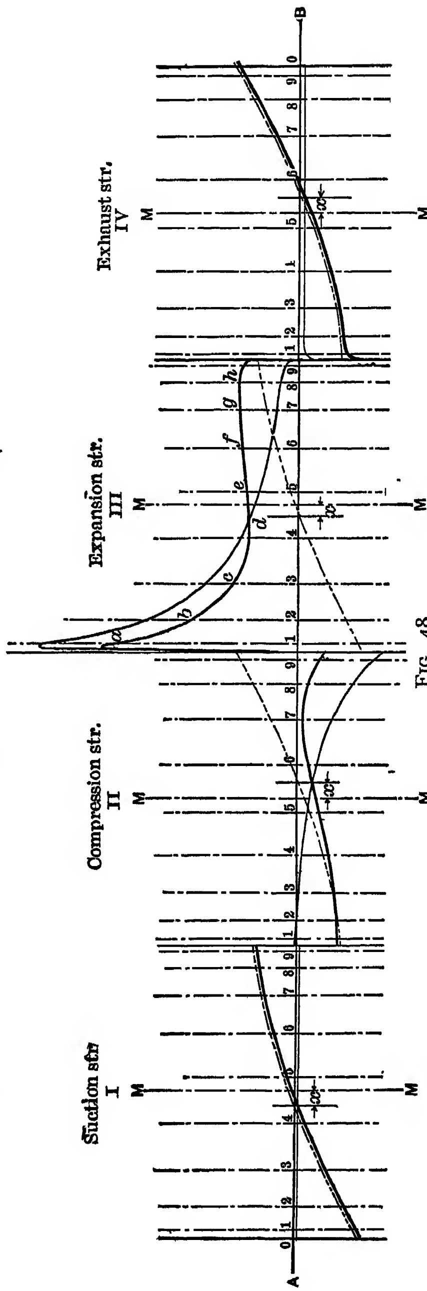

Suction str. I
Compression str. II
Expansion str. III
Exhaust str. IV

FIG. 48.

The curves $C X D$ and $H X K$ are parabolic, but they may, without appreciable error, be drawn as circular arcs.

The Continuous Diagram of the Horizontal Force on the Crank-Pin.—For the sake of better clearness we re-draw the normal diagram representing the pressure in the cylinder during the cycle, and give each stroke of the piston a separate space, so as to obtain a continuous pressure-diagram as shown in light lines in Fig. 48.

Each stroke is subdivided by nine ordinates, the spacing of which corresponds to nine consecutive positions of the crank-pin, spaced 18° apart, and counted from the dead centres. The manner of obtaining the spacing for the ordinates is shown in Fig. 47, and, when transporting it to the four strokes of the diagram, Fig. 48, it will be necessary to note the difference between the spacing toward the head and crank end of each stroke, and place it accordingly.

Offset in Fig. 48 the negative or positive accelerating forces due to the reciprocating parts, at the beginning and at the end of each stroke, observing that the greater positive or negative force is always plotted on the ordinate toward the head end of the stroke. Locate the cutting point X, on the base line A B, 0.08 r from the mid-stroke ordinates M M, toward the head end of the stroke, and draw the curves of force due to the acceleration of the reciprocating parts, as shown in the diagram in broken lines.

The diagram can conveniently be drawn to a scale so as to make the full length of it 12 inches, and to a vertical scale 100 pounds per inch.

We can now combine the continuous pressure-diagram and the curves of force due to the acceleration of the reciprocating parts, by adding, or subtracting, the pressures at the various ordinates, and obtain the curves shown in heavy lines in the diagram. The ordinates under these curves represent the resultant horizontal pressures on the crank-pin. The pressures are positive, representing a promoting force, when they appear above the base-line, and negative, or a resisting force, when they appear below the base-line.

The Tangential Crank-Pin Pressure.—To transform the result-

ant horizontal pressures on the crank-pin into tangential crank-pin pressures proceed as follows:

Draw a circle, *A O*, Fig. 49, representing the crank-circle, and divide the circumference in twenty equidistant points and number them consecutively in accordance with the numbering on the diagram, Fig. 48. Draw radial lines of proper length through points 1, 2, 3, 4, etc., and on each of these offset from the points 1, 2, 3, 4, etc., the lengths of the resultant horizontal crank-pin pressures taken from corresponding points of the diagram, Fig. 48.

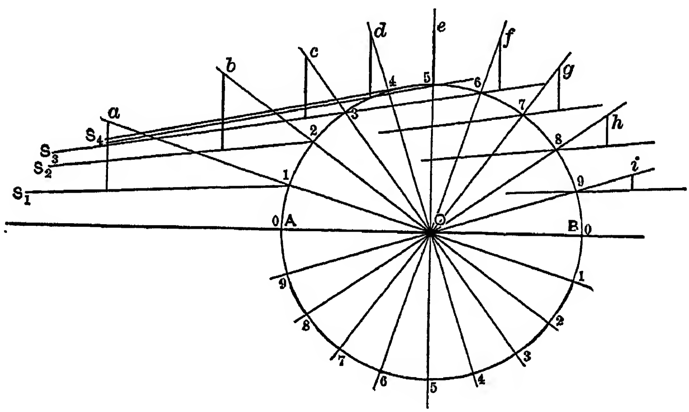


FIG. 49.

Thus we have, for instance, for stroke III of the cycle, the lengths $1-a$, $2-b$, $3-c$, $4-d$, etc., in Fig. 49 plotted to the same length as $1-a$, $2-b$, $3-c$, $4-d$, etc. Fig. 48.

Draw, further, the centre-line of the connecting-rod for each crank-position, as: $1-s_1$, $2-s_2$, $3-s_3$, etc. The lengths of the perpendiculars drawn toward *A B*, from each of the points *a*, *b*, *c*, etc., to the centre-line of the connecting-rod for the corresponding crank-position will be the tangential crank-pin pressures for the positions 1, 2, 3, etc.*

Lay off, in this manner, the length of the tangential crank-pin pressure for all points of the cycle.

The Tangential Cank-Effort Curve.—Let *A B*, Fig. 50,

* For proof of this see page 475 of the Appendix.

represent the length of the crank-pin orbit for one complete cycle. That is, for a four-stroke cycle, $A B = 2 L$ represents $4 \pi r$. Erect on $A B$ four groups of equidistant ordinates, each group containing ten ordinates representing one stroke of the cycle. Number the ordinates in degrees, or in conformity with the numbering of the crank-pin circle, Fig. 47; plot, on each, the tangential crank-pin pressure for the corresponding point, taken from Fig. 49; and through the points thus obtained draw the curve as shown.

This curve is, then, the crank-effort curve, and it shows the intensity and direction of the tangential effort for every position of the crank.

The Areas of Work Performed.—The areas below and above the base-line $A B$ represent the negative, or positive, work transmitted from the fly-wheel to the crank-pin, or from the crank-pin to the fly-wheel of the engine, and the algebraic sum of all the areas represents the total work done during the cycle. This sum should be determined—most conveniently by means of a planimeter—and let it be called A square inches.

The average value of the tangential effort for the cycle, in pounds per square inch piston area, is the ratio

$$\frac{A S_1}{2 L} = T,$$

when S_1 is the vertical scale of the diagram, in pounds per inch.

The force T we mark off on the diagram, above the base-line $A B$, as $A C$, and draw the horizontal line $C D$. The area $A B D C$ represents, then, the total work done during the cycle $= A_1$ foot-pounds per square inch piston area.

The work A_1 can be expressed by the relation

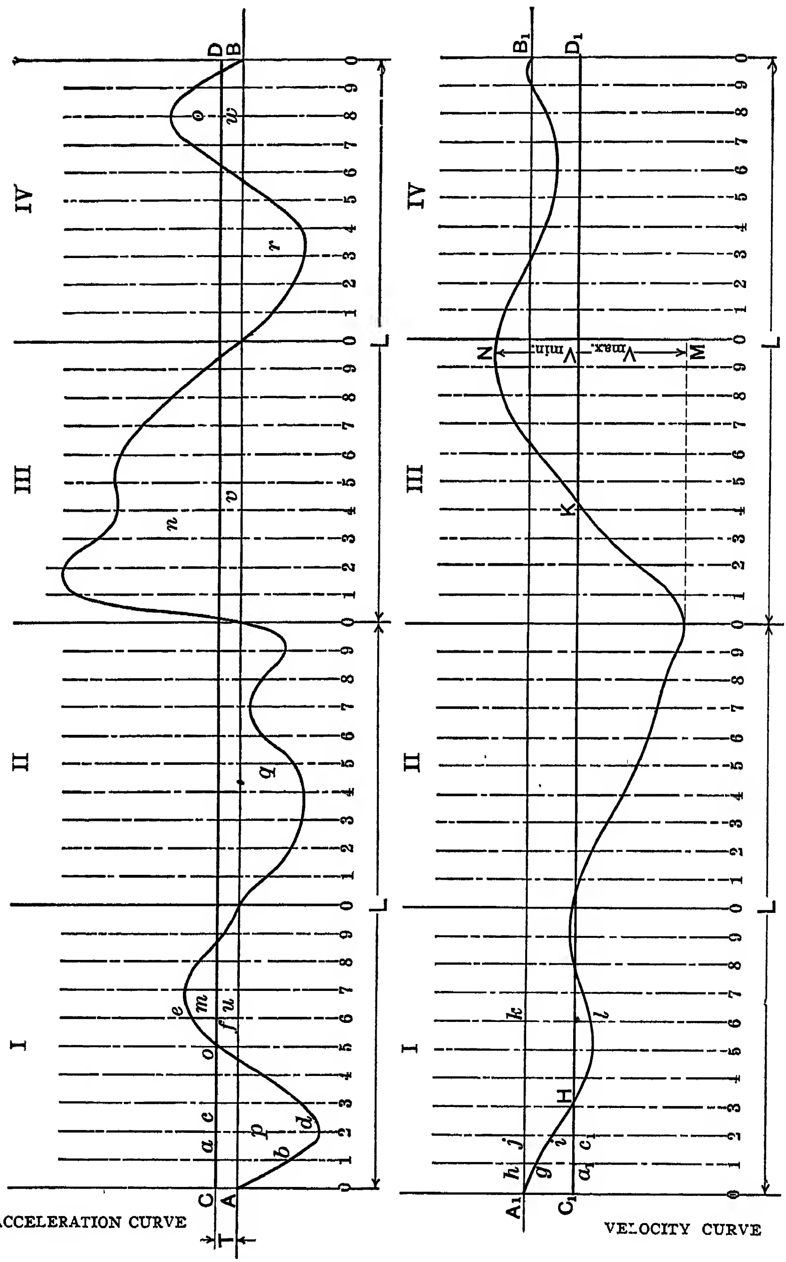
$$A_1 = 4 \pi \frac{r}{12} T,$$

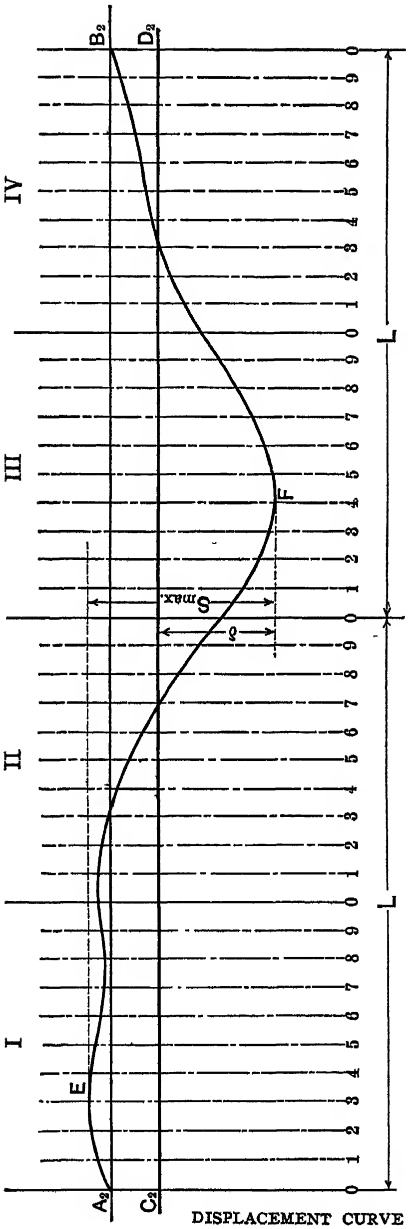
or by
$$A_1 = 2 \frac{r}{12} p_{mc.}$$

when p_{mc} is the mean effective pressure of the indicator-card.

Hence we obtain
$$T = \frac{p_{mc}}{2 \pi} \text{ pounds.}$$

FIG. 50.


ACCELERATION CURVE
VELOCITY CURVE
I
II
III
IV
L
L
L
A
B
C
D
T
A_1
B_1
C_1
D_1
H
K
M
N
V min.
V max.



DISPLACEMENT CURVE

Fig. 52.

The height AC of the diagram should check with this value of T.

Let each of the areas below and above the line $C\ D$ be designated by the letter it contains; the area $n+v$, thus, representing the work done during the expansion-stroke and u, v and w the narrow areas between the base-line $A\ B$ and the line $C\ D$ below the areas m, n and o. We can then write

$$(m+u)+(n+v)+(o+w)+ \\ +[A-(u+v+w)]- \\ -(p+q+r)=A.$$

Hence, $m+n+o=p+q+r$.

That is, according to the diagram, the sum of the areas above the line of normal velocity, $C\ D$, is equal to the sum of the areas below.

That this must be so is self-evident, because the same amount of work that the wheel absorbs, at high velocity, it must deliver when the velocity is below normal.

It is further evident that the largest area of accelerating work, above or below the line of normal velocity, $C\ D$, will cause the wheel to deviate the farthest from its normal speed, and that the weight of

the fly-wheel should, therefore, be determined with respect to this area.

In Fig. 50 the area n is, according to measurements, the area of maximum accelerating work, and its value we assume to be "a" square inches. If L has been made proportionately to the length c of the indicator card, and the vertical scale of the diagram the same as the vertical scale of the indicator card, thus $L = \pi c$, and $S_1 = S$, then the area a represents directly, in terms of the area of the indicator card, the maximum accelerating work in foot-pounds. But as the length L, as well as the vertical scale, may have been chosen arbitrarily, we have generally the area of the maximum accelerating work $= \frac{S_1}{S} \frac{\pi c}{L} a$.

The total work generated during one cycle, per square inch of the piston, is $p_{mc} \frac{2r}{12}$ foot-pounds, and it is represented by the area of the indicator card $= \frac{p_{mc}}{S} c$,

r being the crank-radius, in inches;
p_{mc} the mean effective pressure of the indicator card, in pounds;
S the scale of the indicator-spring; and
$\frac{p_{mc}}{S}$, thus, the mean height of the card, in inches.

If we call the quotient:

$$\frac{\text{The area of maximum accelerating work}}{\text{The area of total work per revolution}} = f$$

we find for a four-cycle single-acting one-cylinder engine

$$f = \frac{\frac{S_1}{S} \frac{\pi c}{L} a}{\frac{1}{2} \frac{p_{mc}}{S} c} = \frac{\pi \frac{a}{L}}{\frac{1}{2} \frac{p_{mc}}{S_1}}. \quad . \quad . \quad . \quad . \quad 103$$

The factor f is called the coefficient for maximum fluctuation of energy, and it is dependent to some extent on the speed of the engine and on the weight of the reciprocating parts, but for engines of similar type and speed it will be found to be in a marked degree constant.

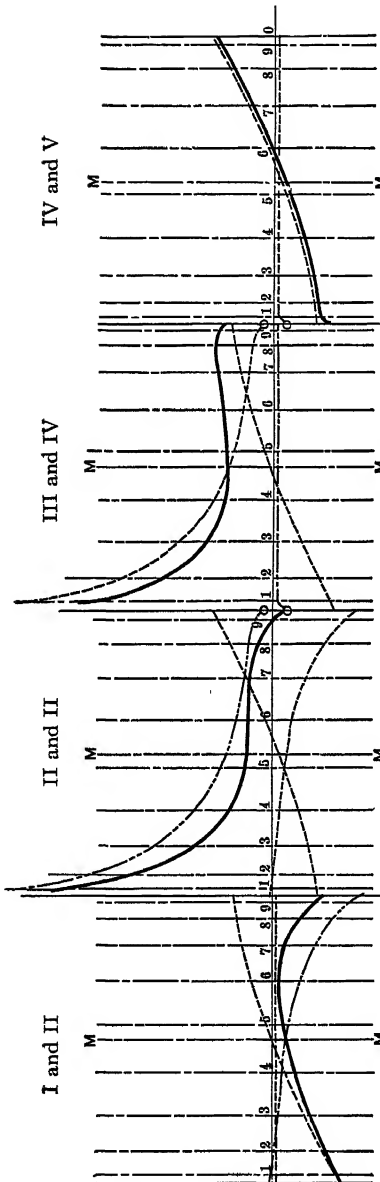


FIG. 53.

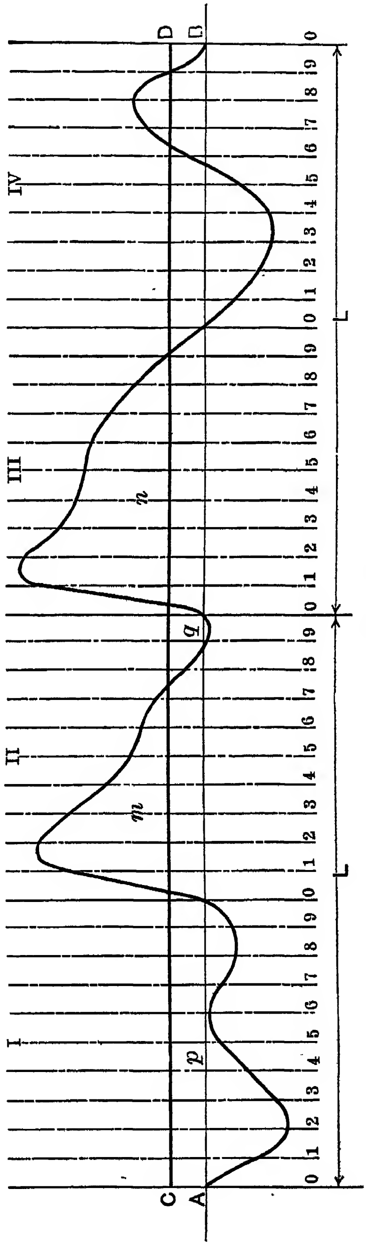


FIG. 54.

Equation 103 is correct when there is only one expansion-stroke for each two revolutions. For double-acting or multiple cylinder engines with 1, 2, or 4 expansion-strokes per revolution the value of a will change somewhat and the denominator will be increased 2-, 4- or 8-fold.

The Coefficient of Maximum Fluctuation of Energy.—Single-Acting, One-Cylinder Engine.—By measurement of the diagram, Fig. 50, we find a = 1.8 square inches, L = 4.71 inches, p_{mc} = 70 pounds, S_1 = 70 pounds per inch, and $\frac{p_{mc}}{S_1}$, thus, = 1.00.

Hence,
$$f = \frac{2 \times 3.14 \times 1.8}{4.71 \times 1.00} = 2.4.$$

Double-Acting, One-Cylinder Engine.—In the double-acting, four-cycle engine, the expansion-strokes follow each other at intervals of 180 and 540 degrees apart, measured by the swing of the crank. It may be thought, therefore, that by displacing two tangential-effort curves for a single-acting engine 180 degrees from each other and combining them we should obtain the effort-curve for the double-acting engine. This is, however, not strictly so, since the force due to the acceleration of the reciprocating parts will be different in the forward stroke from what it is for the return stroke of the piston, due to the influence of the connecting-rod. The effort-curves for the forward and return strokes become, therefore, somewhat different.

In Fig. 53 is shown the continuous diagram of the horizontal force on the crank-pin for a double-acting one-cylinder engine, and from it is obtained the crank-effort curve, Fig. 54.

It will be noticed that, while the area q detracts some from the excess velocity of the wheel, given it by the area m, only a small decrease in velocity will be effected before the area n again effects an increase. It is evident that, in this case, the area $m + n - q$ will be the maximum area of accelerating work.

By measurement we obtain:

The maximum area of accelerating work $m + n - q = a =$ 2.4 square inches, and therefore the coefficient of maximum fluctuation of energy

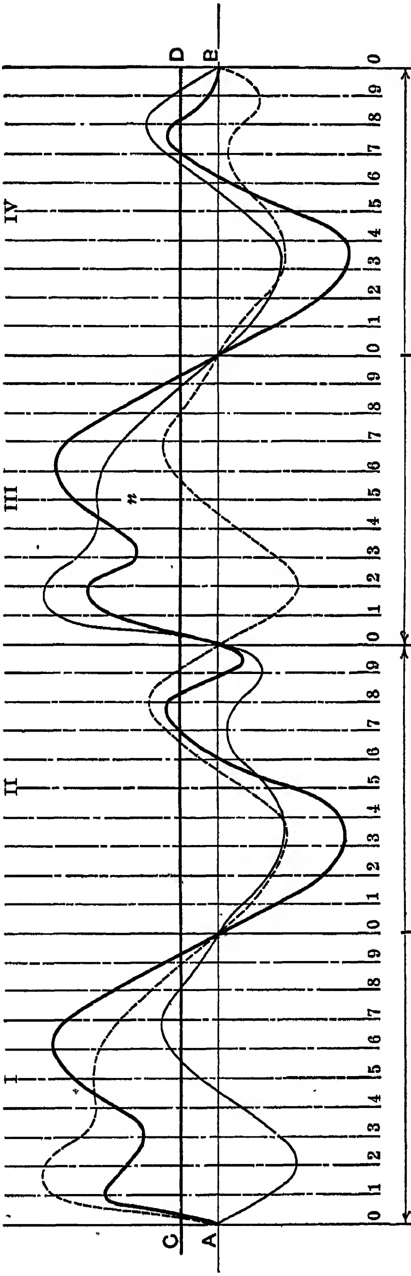


FIG. 55.

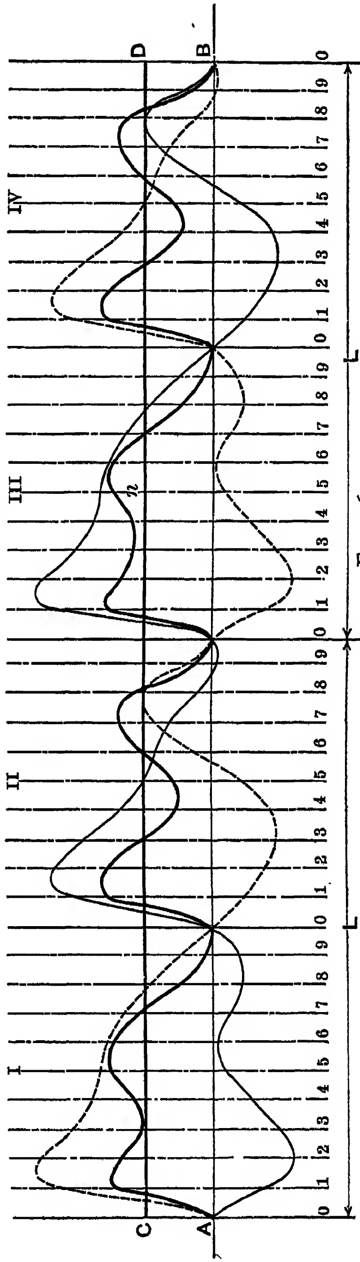


FIG. 56.

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = \frac{3.14 \times 2.4}{4.71 \times 1.00} = 1.6$$

Single-Acting, Two-Cylinder Opposed Engine.—The reciprocating parts will in this type be of the same influence during all expansion-strokes and during similar strokes of the cycle for both cylinders. By displacing two tangential-effort curves for a single-acting engine 180 degrees, and combining them, we obtain, therefore, the effort-curve for the two-cylinder opposed engine. The resulting curve will be practically the same as that shown in Fig. 54, and the coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = 1.6.$$

Single-Acting, Two-Cylinder Twin Engine.—In the two-cylinder twin engine the expansion-strokes follow each other at intervals of 360 degrees, measured by the swing of the crank. The reciprocating parts have the same influence during all expansion-strokes, and the tangential-effort curve can therefore be obtained by combining two curves of a single-acting, single-cylinder engine displaced 360 degrees from each other.

Fig. 55 represents two curves for single engines, which are displaced 360 degrees, and the combined curve is shown in heavy lines.

The area n of maximum accelerating work measures $a = 1.56$ square inches.

The coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = \frac{3.14 \times 1.56}{4.71 \times 1.00} = 1.04.$$

Single-Acting, Two-Cylinder Tandem Engine.—The expansion strokes follow in this type at intervals of 360 degrees, measured

by the swing of the crank, and, as the influence due to the acceleration of the reciprocating parts is practically the same as for two single-acting single engines, the tangential-effort curve can, also in this case, be obtained by the combination of two curves for single engines, displaced 360 degrees from each other.

Fig. 55 is such a curve from which there is obtained as before:

The maximum area of accelerating work $a = 1.56$ square inches. The coefficient of maximum fluctuation of energy

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = 1.04.$$

Twin Single-Acting, Four-Cylinder Opposed Engine.—The expansion-strokes follow each other in this type at every stroke of the engine. To construct the tangential-effort curve we combine the curves for two two-cylinder opposed engines, displaced 180 degrees, and obtain the curve Fig. 56.

The maximum area of accelerating work measures $a = 0.6$ square inches, and the work performed during the cycle is

$$4 \frac{p_{mc}}{S_1} c.$$

Hence, the coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{2 \frac{p_{mc}}{S_1}} = \frac{3.14 \times 0.6}{4.71 \times 2 \times 1.00} = 0.2.$$

Double-Acting, Two-Cylinder Twin Engine.—The tangential-effort curve is obtained by combining two curves for a double-acting single-cylinder engine, as shown in Fig. 56.

The maximum area of accelerating work measures $a = 0.6$ square inches, and the work performed per revolution is $2 \frac{p_{mc}}{S_1} c.$

The coefficient of maximum fluctuation of energy, therefore,

$$f = \frac{\pi \dfrac{a}{L}}{2 \dfrac{p_{mc}}{S_1}} = \frac{3.14 \times 0.6}{4.71 \times 2 \times 1.00} = 0.2.$$

Double-Acting, Two-Cylinder Tandem Engine.—The tangential-effort curve is obtained, also for this type, by combining two curves for a double-acting single-cylinder engine, displaced 180 degrees from each other.

The coefficient of maximum fluctuation of energy becomes, as before,

$$f = \frac{\pi \dfrac{a}{L}}{2 \dfrac{p_{mc}}{S_1}} = 0.2.$$

Twin, Double-Acting Tandem Four-Cylinder Engine.—This type of engine can be arranged with the cranks set at 180 degrees or set at 90 degrees to each other. In the former case double expansion-lines follow each other at intervals of 180 degrees by the swing of the crank, in the latter case one expansion-line occurs every 90 degrees apart.

When the cranks are set at 180 degrees, the effort-curve is obtained by superposing two curves of a two-cylinder double-acting tandem engine, one over the other, and combining them. The result will be an area of maximum accelerating work twice the area for the single curve. The area of the total work will also be doubled, wherefore the coefficient of maximum fluctuation of energy becomes the same as for the two-cylinder tandem,

or $$f = \frac{\pi \dfrac{a}{L}}{4 \dfrac{p_{mc}}{S_1}} = \frac{3.14 \times 1.2}{4.71 \times 4 \times 1.00} = 0.2.$$

When the cranks are set 90 degrees apart, the two curves of a two-cylinder double-acting tandem engine are combined after being displaced 90 degrees from each other, and the area of maximum accelerating work becomes $a = 0.4$.

Hence, the coefficient of maximum fluctuation of energy will be

$$f = \frac{\pi \dfrac{a}{L}}{4 \dfrac{p_{mc}}{S_1}} = \frac{3.14 \times 0.4}{4.71 \times 4 \times 1.00} = 0.07.$$

The deductions of the value of the coefficients f, in the preceding, have all referred to engines of the four-cycle type. The coefficients for the different types of two-cycle engines are practically the same as for the engine of the four-cycle type in which the expansion-strokes succeed each other at identically the same intervals. For instance, in a two-cycle single-cylinder engine, the expansion-strokes occur every revolution, at intervals of 360 degrees by the swing of the crank, the same as in the four-cycle single-acting twin engine. The coefficient for the maximum fluctuation of energy becomes also practically the same as for the latter type, viz.:

$$f = 1.04.$$

It will be evident that the coefficient varies somewhat for different weights of the reciprocating parts, and with the speed of the engine. Its average values for various engine-types are given in the second column of Table XXII, pages 220 and 221.

The Weight of the Fly-Wheel.—By the tangential-effort curves constructed in the preceding, there has been established, for various engine-types, the maximum value of the work generated, at one time or other during the cycle, in excess of that immediately absorbed by the normal resistance. This work, which tends to accelerate the speed of the engine from its minimum to its maximum, is for a four-cycle engine:

$$\text{maximum accelerating work} = 33{,}000\, f \frac{I.H.P.}{N} \text{ foot-pounds;}$$

$\dfrac{I.H.P.}{N}$ being the total work generated per revolution.

The function of the fly-wheel is to absorb the accelerating work, without undue speed-variation. At a change in velocity from V minimum to V maximum, a wheel of the rim-weight W pounds absorbs the work

$$\frac{W}{2g}(V_{max.}^2 - V_{min.}^2) \text{ foot-pounds.}$$

Hence the equation for the maximum change in rim-velocity is

$$\frac{W}{2g}(V_{max.}^2 - V_{min.}^2) = 33{,}000\, f, \frac{I.H.P.}{N}, \quad . \quad (104)$$

when

$I.H.P.$ designates the indicated horse-power generated;
N the number of revolutions per minute;
W the weight of the fly-wheel rim, in pounds, reduced to the mean radius;
R the mean radius of the wheel, in feet;
g the acceleration due to gravity;
$V_{max.} - V_{min.}$ the total change in rim-velocity.

If c feet per minute is the mean velocity of the wheel we have, approximately,

$$c = \frac{V_{max.} + V_{min.}}{2},$$

or we may write

$$c = \frac{2\pi RN}{60},$$

thus

$$(V_{max.} + V_{min.}) = \frac{4\pi RN}{60}.$$

If this value $(V_{max.} + V_{min.})$ be inserted in the main equation, which can be written

$$(V_{max} - V_{min.})\ (V_{max.} + V_{min.}) = 2 \times 33{,}000\, g \frac{f\, I.H.P.}{NW},$$

we obtain

$$V_{max.} - V_{min.} = \frac{33{,}000\ g\ 60}{2\pi} \frac{f\, I.H.P.}{RN^2W}, \quad . \quad (105)$$

and

$$\frac{V_{max.} - V_{min.}}{c} = \frac{33{,}000\ g\ 60^{-2}}{4\pi^2} \frac{f\, I.H.P.}{R^2N^3W}.$$

The quantity $\frac{V_{max.} - V_{min.}}{c}$, is an expression for the steadiness in the velocity of the wheel, which may be made anything desired. Let this factor be designated by the coefficient $\frac{1}{K}$, and let the numerical factor $\frac{33{,}000\ g\ 60^{-2}}{4\pi^2}$ be substituted by its approximate value, 96,400,000.

The formula, becomes, then

$$W = K\frac{96{,}400{,}000\, f\, I.H.P.}{R^2\, N^3}. \quad . \quad . \quad . \quad (106)$$

For single-acting, one-cylinder engines the previous formula may be made more convenient by fixing in advance on an allowable rim speed S. $R^2\, N^2$ becomes then $= \frac{S^2}{40}$ and the formula may be written

$$W = \frac{C\ I.H.P.}{N}, \quad . \quad . \quad . \quad . \quad . \quad (106a)$$

$I.H.P.$ being the indicated power of the engine,
N the number of revolutions per minute, and
C a coefficient varying with the rim-speed.

Value of the Coefficient C.

Rim Speed (of C. of Gravity of Rim).	3500 Ft.			4000 Ft.			4500 Ft.			5000 Ft.		
	Standard Wheel.	Medium Heavy Wheel.	For Electric Light Engines.	Standard Wheel.	Medium Heavy Wheel.	For Electric Light Engines.	Standard Wheel.	Medium Heavy Wheel.	For Electric Light Engines.	Standard Wheel.	Medium Heavy Wheel.	For Electric Light Engines.
K	30	50	70	30	50	70	30	50	70	30	50	70
C	22700	27800	53000	17300	29000	40000	13700	22000	32000	11000	18000	26000

Table XXI, page 218, gives values for K suitable for various services.

If the rim speed be figured at the periphery of the wheel, then, assuming the outside radius of the wheel, R_o, to be 1.1 of its mean radius, R, the coefficient, C becomes approximately 20 per cent larger than the figure given in the table.

The Acceleration Curve for the Revolving Weights.—The driving effort acting tangentially on the crank-pin is represented graphically for all positions of the crank by the crank-effort curve,

Fig. 50, but by multiplying this force, measured at any point of the cycle, by the quantity $\frac{W}{g}$ we obtain the acceleration which the force gives a weight W. The curve, Fig. 50, can, therefore, be said also to be the acceleration curve for the revolving weight W.

The Velocity Curve.—Let it be assumed that the distance $A\ B$, Fig. 50, instead of representing the length of the crank-pin orbit for two turns of the crank, represents the time it takes the wheel to make two revolutions. This assumption we can make, because the difference between the intervals of time and the intervals of space is so small that it would hardly be measurable in the diagram.

The distance between each two ordinates of the diagram represents, then, the time for $\frac{1}{20}$ of one turn, and the length of the mean ordinate between the base-line and the acceleration curve during the interval represents the average of the variable acceleration a weight W is given during the elementary time $\frac{1}{20}\ \frac{60}{N}$; $\frac{60}{N}$ being the time for one revolution.

The increase in velocity of a moving object during a time-element is the product of its acceleration and the time-element;

$$\text{Velocity} = \text{Acceleration} \times \text{Time}.$$

Hence we see, that the area of each figure, such as $b\ a\ c\ d$, enclosed by the base-line $C\ D$, the acceleration curve and two ordinates represents the increase in velocity attained during the time represented by the distance between the ordinates.

In order to represent graphically the increase or decrease in velocity of the revolving masses during the cycle, we may, therefore, integrate the elementary velocity-areas such as $A\ C\ a\ b$, $b\ a\ c\ d$, etc., from the time of the beginning of the cycle at A until its end at B, plot their sum on corresponding ordinates with line $A_1\ B_1$, Fig. 51, as base, and draw a curve through the points thus obtained.

For instance, the line $g\ h$ is plotted of a length so as to represent the area $A\ C\ a\ b$ and the length $i\ j$ to represent the area $A\ C\ c\ d$,

and so forth. The increase in velocity is positive when the velocity-area is located above the base-line $C\,D$ and negative when located below. Hence, the length $k\,l$ represents the area $C\,A\,d\,O - O\,e\,f$; the area $C\,A\,d\,O$ being negative, and the area $O\,e\,f$ positive.

The vertical distance between each minimum point and the next maximum point of the curve is the measure of the increase in velocity which the revolving masses have attained during the corresponding time, and if n is the area of maximum accelerating work, the distance $N\,M$ must be the maximum change in velocity that has occurred during the cycle $= V_{max.} - V_{min.}$

The Displacement Curve.—The line $C_1\,D_1$ is the line of normal velocity, drawn so as to make the sum of the areas above the line of the same value as the areas below.

The areas, such as $g\,i\,c_1\,a_1$, between the line $C_1\,D_1$, the velocity-curve and any two ordinates are the product of the variable excess or deficiency in velocity and the time-element; and as space also is a product of velocity and a time-element,

$$\text{Space} = \text{Velocity} \times \text{Time},$$

we may integrate the areas between the velocity-curve and the line of normal velocity $C_1\,D_1$, in the same manner the velocity-areas were integrated, plot their sum on corresponding ordinates with $A_2\,B_2$ as base, and obtain the displacement curve $A_2\,E\,F\,B_2$, Fig. 52.

The ordinates of this curve represent the distance which a point on the fly-wheel, or any point of the revolving system, is, at any time, ahead or behind the position it would at the time have gained with a perfectly uniform velocity.

The point F, following a period of low velocity, is a minimum point—a generator pole in the revolving system there being a maximum distance behind, and E, following a period of high velocity, is a maximum point, where the pole is a maximum distance ahead of its position due to uniform velocity.

The Weight of the Fly-Wheel for a Limited Pole-Displacement.—The areas $K\,N\,B_1\,D_1 + A_1\,H\,C_1$, Fig. 51, represent the total displacement of a pole, counted from the minimum-point to

the following maximum-point; and in the diagram, which is a curve for a single-cylinder engine, the sum of these areas measures

$$\frac{1}{12}(V_{max.} - V_{min.}) \times A_1 B_1.$$

Hence, as $A_1 B_1$ represents $\frac{2 \times 60}{N}$ seconds, we have

$$S_{max.} = \frac{1}{12}(V_{max.} - V_{min.})\frac{120}{N} \text{ feet.}$$

It may be assumed, referring to Fig. 52, that, on an average,

$$\delta = \tfrac{3}{5} S_{max.}$$

That is, we may assume that the fly-wheel deviates three-fifths of the total distance $S_{max.}$ toward one side of the position of normal speed, and two-fifths toward the other.

The maximum deviation of a pole from the position due to uniform speed we get

$$\delta = \frac{6}{N}(V_{max.} - V_{min.}). \quad . \quad . \quad . \quad (107)$$

This value combined with equation 105 gives for a single-acting single-cylinder engine

$$W = 10{,}000{,}000 \frac{6.054 f \ I.H.P.}{R N^3 \delta}. \quad . \quad . \quad . \quad (108)$$

For multiple-cylinder and double-acting engines the numerical factor, 6, in the preceding equation 107, will generally be a somewhat smaller value, but it may be assumed that, under the most unfavorable conditions in each case, δ varies proportionately with $V_{max.} - V_{min.}$, and the approximation expressed by equation 107 will then apply in all cases.

If the value $6.054 f$ be designated by a new coefficient, F_1, we obtain the general equation for all engine types:

$$W = 10{,}000{,}000 \frac{F_1 \ I.H.P.}{R N^3 \delta}. \quad . \quad . \quad . \quad . \quad (108a)$$

As

$$\delta = \frac{2 \pi R \gamma}{360}$$

when γ is the angle of deviation of a point in the revolving system,

therefore,

$$W = 10{,}000{,}000 \frac{F_2 \ I.H.P.}{R^2 N^3 \gamma}. \quad . \quad . \quad . \quad . \quad (108b)$$

If ϵ is the fluctuation in degrees phase, and P the number of poles in the generator, we have

$$\gamma = \frac{2}{P}\epsilon,$$

and, hence, also $$W = 10{,}000{,}000 \frac{F_3}{R^2} \frac{I.H.P.}{N^3} \frac{P}{\epsilon}. \quad . \quad (108c)$$

The values of the coefficients F_1, F_2, and F_3 are:

$$F_1 = 6.054f, \quad F_2 = \frac{360}{2\pi} 6.054 f, \quad \text{and} \quad F_3 = \frac{360}{4\pi} 6.054 f;$$

and their numerical values for different engine types, expressed in round numbers, are given in Table XXII.

When the number of poles in the generator is small, equation 106 becomes of higher value than equation 108*c*, and the wheel should then be determined with respect to its fluctuation in velocity.

By equating the formulas 106 and 108*c* we obtain $P = 0.0555\ \epsilon K$;

and hence, when $\epsilon = 2.5°$ and $K = 100$, then $P = 13.87$;
and when $\epsilon = 2.5°$ and $K = 150$, then $P = 20.80$.

The minimum number of poles for which, accordingly, the pole-displacement should be made the basis for the computation of the fly-wheel weight required for parallel operation of alternators is:

14 poles, when K is required to be 100,
and 20 poles, when K is required to be 150;
$\epsilon = 2.5°$ being assumed to be a minimum value used in practice.

The Fly-Wheel Formulas.—The weight of the fly-wheel should be determined with respect to the degree of uniformity of rotation that will be required for the special service for which an engine is intended.

As the necessity for extreme steadiness of rotation and heavy wheels varies materially for different services, and as the type of the engine has a particular influence on the steadiness a given wheel will impart to the speed, the formula by which the wheel is computed should, properly, include one factor varying with the

engine-type and one factor varying with the degree of steadiness called for.

A general formula filling these requirements, and which is correct within a very slight approximation, reads:

$$W = \frac{C f \ I.H.P. \ K}{R^2 N^3}; \quad . \quad . \quad . \quad . \quad . \qquad (106)$$

C is a constant $= \dfrac{60^{-2} \times g \times 33{,}000}{4\pi^2} = 96{,}400{,}000$, approximately;

g being the acceleration due to gravity.

W is the weight of the wheel reduced to the radius R;

$I.H.P.$ the maximum indicated horse-power of the engine,

R the mean radius of the wheel rim, in feet,

N the number of turns per minute,

f the coefficient of maximum fluctuation of energy

$$= \frac{\text{Maximum accelerating energy}}{\text{Total energy developed during one revolution}},$$

K is a coefficient for the allowable fluctuation in speed, expressed by the ratio of the normal speed to the maximum speed-variation;

thus $$\frac{1}{K} = \frac{\text{maximum speed} - \text{minimum speed}}{\text{normal speed}}.$$

The values of K and f may conveniently be selected from Tables XXI and XXII to suit the requirements in any special case.

TABLE XXI.

Values of Coefficient K Suitable for Various Services.

	K
For ordinary industrial purposes, belt drive	25– 35
For small electric light installations, direct current, belt-driven	50– 60
For pumping machinery direct connected to engine	60–100
For large electric light installations, direct current, belt-driven	60– 80
For large electric light installations, direct current, direct connected	90–120
For gear-wheel transmissions	90–120
For blast-engines	90–150

By formula 106 the fly-wheel is determined only with respect to its fluctuations in angular velocity. The necessity that alternating current generators, working in parallel, should rotate with

as nearly perfect synchronism as possible demands, however, a regulation of the engine, not based on speed fluctuations, but based on the maximum pole-displacement that may be allowed. By pole-displacement is meant the deviation of a pole, due to variable speed, in advance and in retard of a point revolving with it with the same mean, but perfectly uniform, speed.

The following formula determines the required weight of the revolving masses, in order to keep the angular deviations of the generator-poles inside a predetermined limit γ:

$$W = 10{,}000{,}000 \frac{F_2 \ I.H.P.}{R^2 N^3 \gamma}. \quad . \quad . \quad . \qquad (108b)$$

Builders of alternating-current generators specify, generally, that the uniformity of operation must be such as to limit the deviation of a pole, on either side of the position of absolute uniform speed, to a certain given number of degrees phase, often $2\frac{1}{2}°$ to $3°$.

The distance between two poles of similar polarity being counted 360 degrees phase, or the distance between two consecutive poles 180 degrees, the relation between the angular degrees, measured on the pitch-circle of the generator poles, and the number of degrees phase will be the following:

$$\gamma = \frac{2}{P}\epsilon,$$

γ being the angular degrees,
ϵ the number degrees phase, and
P the number of poles.

Formula 108*b* can, therefore, for convenience be written

$$W = 10{,}000{,}000 \frac{F_3 \ I.H.P. \ P}{R^2 N^3 \epsilon}. \quad . \quad . \quad . \qquad (108c)$$

Sometimes the number of poles in a generator is not specified directly, but, instead, the number of cycles and number of revolutions are given. The number of poles can then be ascertained by the formula,

$$P = \frac{120 \times \text{number of cycles}}{\text{number of revolutions}}.$$

TABLE XXII.

Values of Coefficients f, F_1, F_2 and F_3 for Various Engine Types.

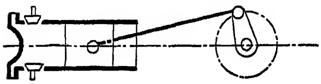
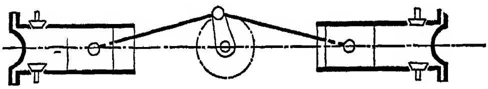
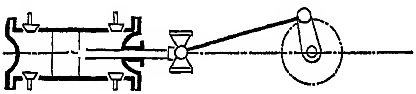
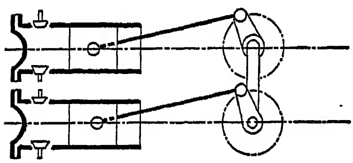
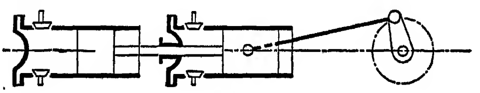
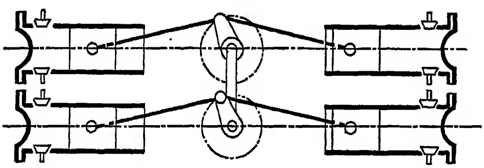
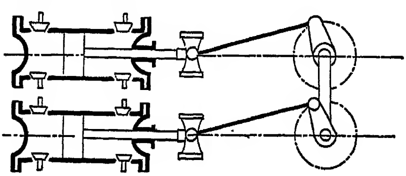
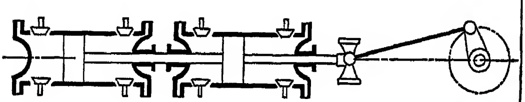
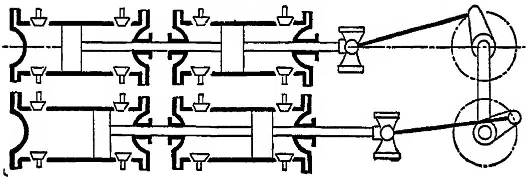
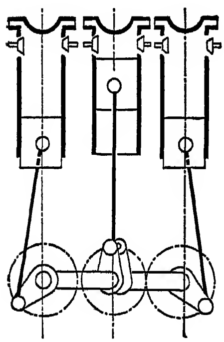
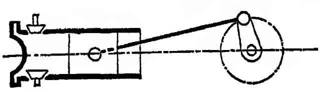
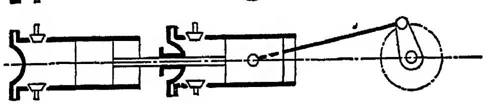
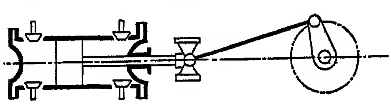
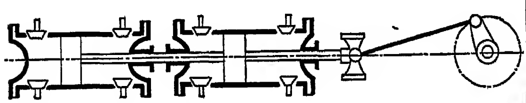
Cylinder Arrangement.	Type.	f	F_1	F_2	F_3
Four-Cycle Engines.					
	I	2.4	14	800	400
	II	1.6	10	560	280
	III	1.6	10	560	280
	IV	1.04	6.4	360	180
	V	1.04	6.4	360	180
	VI	0.2	1.2	70	35
	VII	0.2	1.2	70	35
	VIII	0.2	1.2	70	35

TABLE XXII—*Continued.*

Values of Coefficients f, F_1, F_2 and F_3 for Various Engine Types.

Cylinder Arrangement.	Type.	f	F_1	F_2	F_3
Four-Cycle Engines.					
	IX	0.07	0.44	24	12
	X	0.6	3.6	210	105
Four-cylinder engine, same type as above; cranks 90 degrees apart.	XI	0.2	1.2	70	35
Two-Cycle Engines.					
	XII	1.04	6.4	360	180
	XIII	1.04	6.4	360	180
	XIV	0.3	1.8	104	52
	XV	0.07	0.44	24	12

Example.—What diameter, and weight, of the wheel would be suitable for an 18 × 28 four-cycle, single-cylinder engine running at 190 turns per minute on suction producer-gas, the engine to be belted to a direct-current generator?

The engine will develop 90 B.H.P. on suction producer-gas, corresponding to 106 I.H.P.

The rim speed we could allow at 5,400 feet per minute, giving a wheel-diameter of 9 feet.

The required width of the belt will be, according to formula 128, in the appendix, $W = \frac{9 \; B.H.P.}{V \; 0.9} = \frac{9 \times 90}{90 \times 0.9} = 10$ inches; assuming the belt to run over a small driven pulley that gives an arc of contact of about 150°.

In order to provide a wheel of ample width, including space for a set of ratchet teeth that will be required alongside one edge, the total width of the wheel-face may finally be approximately 18 inches. If the thickness of the rim be made 10 inches, the mean radius of the wheel-rim becomes $R = 4' - 1'' = 4.09$ feet.

The coefficient K, of equation 106, would in this case, according to Table XXI, be selected between the value $K = 60$ and $K = 80$, and we choose the middle value $K = 70$.

Inserting, then, the various numerical values in formula 106 we get

$$W = \frac{96,400,000 \times 2.4 \times 106 \times 70}{16.7 \times 6,860,000}$$

$$W = 15,000 \text{ pounds.}$$

For a twin engine of the same size cylinders as the above engine the required weight of the wheel, to give the same steadiness, would be

$$W = \frac{1.04 \times 2}{2.4} \; 15,000$$

$$W = 13,000 \text{ pounds.}$$

The proper weight of the fly-wheel for single-acting single-cylinder engines of the above dimensions, if intended for the operation of alternators in parallel, may be obtained by equation 108*c*.

Assuming the number of poles in the generator to be 14, and the maximum pole-displacement required to be inside 2½ degrees phase,

$$\text{We have } W = 10{,}000{,}000 \frac{F_3 \ I.H.P.}{R^2 N^3} \frac{P}{\epsilon},$$

$$\text{hence,} \quad W = 10{,}000{,}000 \frac{400 \times 106}{16.7 \times 6{,}860{,}000} \frac{14}{2.5},$$

$$\text{or,} \quad W = 21{,}000 \text{ pounds, approximately.}$$

The same result will, in the case of 14 poles, be obtained by equation 106, using a coefficient $K = 100$.

The above weight includes the effective fly-wheel weight of the generator-armature; that is, its weight reduced to the radius R.

In case the number of poles were 56 instead of 14, the required total fly-wheel weight, reduced to the radius R, becomes 83,000 pounds.

A single-acting, two-cylinder opposed engine, or a double-acting, one-cylinder engine, types II and III, would require, for 56 poles, a fly-wheel weight approximately 60,000 pounds, and a single-acting twin or tandem engine, types IV and V, requires only 37,000 pounds.

It is evident, therefore, that for reducing the fly-wheel weight, when there are rigid requirements in respect to pole-displacement, the engine types IV and V give far the better results than the types I, II or III.

Example.—As an illustration from an actual case may be quoted the engines of the A. B. Dick Co., of Chicago, Ill., which furnish electric current for the lighting of the building and for general motive power in the factory. The engines are: one single unit 14¾ × 24, rated at 55 B.H.P., and one twin engine of the same cylinder dimensions, rated at 110 B.H.P. The maximum indicated horse-power is, on producer gas, respectively, 70 and 140. Both units are direct connected to direct-current generators, and run 200 revolutions per minute.

The fly-wheels are 8′ – 6″ outside diameter and their mean radius of the rim is 3.9 feet.

The weight of the wheel for the twin-engine is	11,500 lbs.
Its weight reduced to the mean rad. of the rim	9,900 lbs.
The armature wgt. red'd. to the mean rad. of the rim	1,600 lbs.
Total wgt. red'd. to the mean rad. of the rim	11,500 lbs.
The corresponding weight for the single engine is	12,100 lbs.

Inserting the various values in formula 106, using, in the case of the twin engine, $f = 1.04$ and, in the case of the single engine, $f = 2.4$, and solving for K we obtain:

In the case of the twin engine, $K = 100$,
and in the case of the single engine $K = 90$.

The current furnished by these engines produces, under all conditions, a very satisfactory light, both when the engines run singly or in parallel. The values of the coefficient K, from 90 to 120, as quoted in Table XXI, for services such as the above, can, therefore, be considered fully conservative.

EXAMPLE.—The installation of three "Snow" four-cycle double-acting, twin-tandem engines at the Gas and Electric Light Co.'s station in San Francisco, which operate alternating-current generators very successfully in parallel, may be used as an illustration for the computation of heavy wheels.

The engines are of the following general specifications:

Four double-acting cylinders, 42×60. Power, 4,000 rated B.H.P., or 5,400 maximum I.H.P. The fly-wheel, 23 feet outside diameter; mean radius 11 feet. Its weight, 97,000 pounds, and it makes 88 revolutions per minute.

The generators are 25 cycle—34 poles.

For the above case we use equation 108*c*.

$$W = 10{,}000{,}000 \frac{F_3\ I.H.P.}{R^2\ N^3}\ \frac{P}{\epsilon}.$$

Inserting the various values, of which we obtain from Table XXII, for a twin-tandem, double-acting engine, $F_3 = 12$, and assume that the allowed maximum fluctuation in degrees phase is $2\frac{1}{2}°$, we get

$$W = 10{,}000{,}000 \frac{12\quad 5{,}400}{121\ 680{,}000}\ \frac{34}{2.5} = 106{,}000 \text{ pounds.}$$

The actual weight of the wheel is	97,000 lbs.
Its weight reduced to the mean rad. of the rim	80,000 lbs.
The weight of the generator armature, reduced to the mean rad. of the rim, approximately	25,000 lbs.
Total weight reduced to the mean rad. of the rim, approximately, ...	105,000 lbs.

Had the number of poles of the generator been 20, instead of 34, the required weight of the wheel would become

$$W = 64{,}000 \text{ pounds.}$$

Approximately the same weight is obtained by equation 106 if the value $K = 150$ is used.

Heavy wheels, such as referred to in the above example, are in practice generally determined with guidance from displacement curves constructed as explained in the preceding, but it becomes quite possible to expedite preliminary determinations by the use of formula 108*c*, which, with the coefficients quoted in Table XXII, gives acceptable results.

EXAMPLE.—As a further application of the fly-wheel formulas, a determination may be made of the weight of wheels required for 44 × 54 double-acting twin tandem engines operating on blast-furnace gas, and direct-connected to 36-pole alternating-current generators. The alternators to operate in parallel at a speed of 83⅓ revolutions per minute, giving, consequently, 25 cycles.

Assuming the gas to be of such quality as to give a mean effective pressure of 68 pounds, and that the piston-rod is 12 inches in diameter, then the maximum output of the engine will be 4,300 I.H.P.

If the coefficient of the phase-displacement at a maximum output $\epsilon = 3°$ be allowed, and assuming the mean diameter of the wheel to be 21.2 feet, then the required weight of the wheel, reduced to the mean diameter, according to equation 108*c*, will be

$$W = 10{,}000{,}000 \frac{12 \times 4{,}300 \times 36}{112 \times 578{,}000 \times 3} = 92{,}500 \text{ pounds.}$$

The above data are those determining the wheels of the Allis-

Chalmers engines operating the alternating current generators at the plant of the Indiana Steel Co., at Gary, Ind.

The wheels of these engines are 23 feet outside diameter and have a rim-section 16 inches wide at the face by 21 inches deep.

The rim of the wheel weighs	68,000 lbs.
The arms and hub, approximately	23,000 lbs.
The total weight of the wheel	91,000 lbs.
The weight of arms and hub, reduced to the mean rim-dia. is .	7,000 lbs.
Total weight of one wheel, reduced to the mean rim-dia.	75,000 lbs.
The weight of the revolving generator-field, reduced to the mean wheel-dia., approximately	20,000 lbs.
Total fly-wheel wgt. reduced to the mean wheel-dia., 21.2 ft. .	95,000 lbs.

The normal capacity of the alternators is 2,000 K.W., but they have a guaranteed overload capacity of 30 per cent. At normal load, which accordingly is in the neighborhood of 3,200 I.H.P., the phase-displacement will be, approximately, $\epsilon = 2\frac{1}{2}°$.

CHAPTER X

THE CRANK-SHAFT

In the European gas-engine practice, the custom has been general to make the engine shaft of a centre-crank type. This construction is very convenient and safe for self-contained engines, and must be considered the standard for engines of small and medium size. When, however, engines are connected in pairs, as twin engines, carrying on the shaft between them heavy wheels and generator armatures, there will, with this construction, be required to place five or more main bearings in line for the proper support of the shaft. This arrangement is by no means desirable, and will not be safe, considering the liability of the alignment of the bearings to get out of true. A better arrangement with respect to large engines, particularly of the twin type, is to use shafts with side-cranks and only two main bearings, even though it becomes necessary to make the shaft of a considerably larger diameter.

Roughly, it may be said, that a side-crank shaft will be of a diameter 50 per cent larger than one with centre-crank for the same cylinder capacity. But, on the other hand, it may be made shorter, and a saving in weight is also often effected, as well as sound material insured, by forging the shaft hollow.

The principal points to be considered with respect to the design of the crank-shaft are: its strength and rigidity with reference to bending and torsional forces, and the proper bearing surfaces of its journals.

Forces Acting on the Crank-Pin.—The maximum pressure on the piston, at the time of the explosion in the cylinder, varies from, approximately, 350 pounds per square inch in engines of low compression to 450 pounds in producer-gas engines with high compression. The former figure could safely be used as basis for the computation of the shaft for gasoline or illuminating-gas engines, whereas 450 pounds would be safer with respect to producer-gas or blast-furnace gas engines.

If the maximum pressure per square inch of piston is, p, pounds, and the area of the piston, A, square inches, the maximum total pressure on the piston becomes $P = pA$. This pressure is transmitted directly to the crank-pin, when the crank passes the centre at a slow speed.

In Fig. 57 is reproduced the tangential-effort curve for the pressure stroke of a single-cylinder engine. The highest tangential pressure occurs at the point x, approximately 30 degrees from the beginning of the explosion stroke by the swing of the crank. At a corresponding distance from the head-end of the

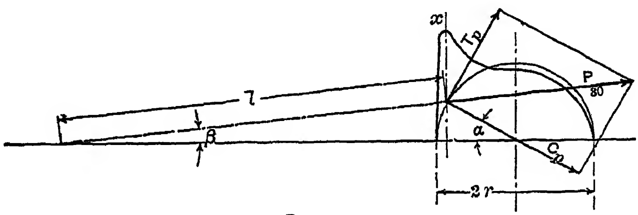


Fig. 57.

indicator-diagram the total pressure on the piston measures, approximately, 75 per cent of the total initial pressure P.

The elevation of the crank, above the centre-line of the engine, at this point being small, the pressure on the piston is practically the same as the pressure in the direction of the connecting-rod.

The pressure, P_{30}, on the crank-pin is, therefore, approximately, 75 per cent of the explosion pressure P,

$$P_{30} = 0.75\,P.$$

According to the diagram, we have

$$\sin\beta = \frac{r \sin\alpha}{l},$$

or for the average value $\frac{r}{l} = \frac{1}{5.5}$, and $\alpha = 30°$,

$$\sin\beta = 0.09$$

and $$\beta = 5° - 10'.$$

Hence, $$\cos(\alpha + \beta) = \cos(35° - 10') = 0.817.$$

$$\sin(\alpha + \beta) = \sin(35° - 10') = 0.576.$$

The tangential and radial forces acting on the crank-pin will, therefore be

$$T_p = sin\ (\alpha + \beta)\ P_{30} = 0.576\ P_{30}$$
$$C_p = cos\ (\alpha + \beta)\ P_{30} = 0.817\ P_{30}.$$

The strength of certain portions of the shaft is taxed most by the force, P, prevailing when the crank passes the centre, while

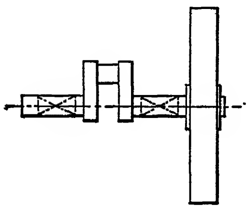


Fig. 58.

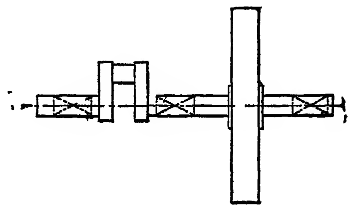


Fig. 59.

other portions are taxed most by the force, P_{30}, prevailing when the crank stands 30 degrees from the centre. It will, therefore, be necessary to analyze the strength of the shaft with reference to both of these forces.

Small- and medium-sized engines have often the fly-wheel carried on the end of the shaft overhanging the main journal, as

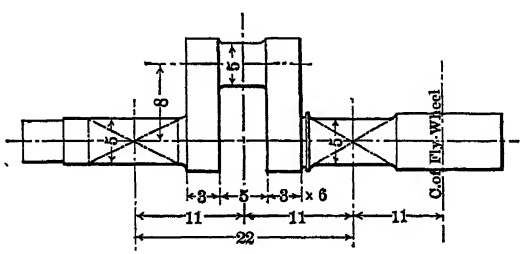


Fig. 60.

Fig. 58. Large engines should always have a third bearing for the support of the end of the shaft, as Fig. 59, in order to avoid the heavy strains in the shaft and crank, due to the weight of the wheel, that otherwise would be set up.

The following two examples explaining the method generally

employed for computing the strength of the various parts of centre-crank shafts relate, the first one, to a shaft carried by two bearings and, the second, to a shaft carried by three bearings.

The Strength of Centre-Crank Shafts Supported by Two Bearings.—Fig. 60 is a preliminary sketch for a shaft intended for a 9 × 16 horizontal producer-gas engine to run at 230 R.P.M. and it will be required to analyze the stresses that will obtain in the various parts.

The area of a 9-inch piston is 63.6 square inches, which for a pressure of 450 pounds per square inch gives a total pressure on the piston $P = 28{,}600$, approximately. The power of the engine will be 17 B.H.P., or 20 I.H.P., and a wheel weighing 3,000 pounds and of 6 feet diameter will give a satisfactory steadiness of rotation (the coefficient of steadiness being $K = 50$).

The Crank on Centre.—In Fig. 61 are represented diagrammatically the forces acting on the shaft, when the crank, at the time of the explosion in the cylinder, passes the head-end centre.

P is the total pressure on the piston, and H_1 and H_2 the reactions on the bearings due to this force. W is the weight of the wheel and V_1 and V_2 the reactions in the bearings due to it.

The symbols, c_1, c_2 and f, denote, according to the figure, the distances between the left-hand and right-hand bearings and the centre of the engine, and between the right-hand bearing and the centre of the fly-wheel; n is the distance, centre to centre, between the bearings.

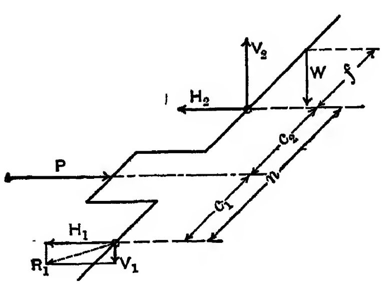


Fig. 61.

The data given are:

$$c_1 = c_2 = 11''$$
$$f = 11''$$
$$n = 22''$$

$$P = 28{,}600 \text{ pounds.}$$
$$W = 3{,}000 \text{ pounds.}$$

We obtain

$$H_1 = \frac{c_2}{n} P = \frac{11}{22} 28{,}600 = 14{,}300 \text{ pounds.}$$

$$H_2 = \frac{c_1}{n} P = \frac{11}{22} 28{,}600 = 14{,}300 \text{ pounds.}$$

$$V_1 = \frac{f}{n} W = \frac{11}{22} 3{,}000 = 1{,}500 \text{ pounds.}$$

$$V_2 = \frac{n+f}{n} W = \frac{33}{22} 3{,}000 = 4{,}500 \text{ pounds.}$$

$$R_1 = \sqrt{H_1^2 + V_1^2} = \sqrt{14{,}300^2 + 1{,}500^2} = 14{,}380 \text{ pounds.}$$

In Fig. 62 are represented the forces that strain the material (the same forces as in Fig. 61), and also the sections of the material offering resistance, at the various points of the shaft.

Additional data given are:

$$d = d_1 = d_2 = 5'',$$
$$c_1 - e = 7'',$$
$$b = 3'', \quad h = 6''.$$
$$e = 4'', \quad r = 8''.$$

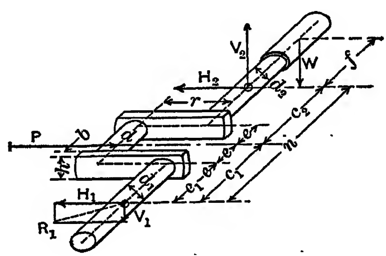


FIG. 62.

Maximum Strain at the Middle Section of Crank-Pin.—The bending moment is

$$M_b = R_1 c_1 = 14{,}380 \times 11 = 158{,}180.$$

$$\text{Section modulus} = \frac{J}{a}* = 0.1\, d^3 = 0.1 \times 5^3 = 12.5.$$

Maximum bending strain in pin

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{158{,}180}{12.5} = 12{,}650 \text{ pounds.}$$

The shearing strain at the ends of pin is

$$S_s = \frac{2\,P}{\pi\, d^2} = \frac{2 \times 28{,}600}{3.14 \times 25} = 733 \text{ pounds.}$$

* Section Modulus, $\frac{J}{a} = \frac{\text{Mom. of Inertia}}{\frac{d}{2}} = \frac{\pi}{32} d^3 = \text{appr. } 0.1\, d^3.$

The twisting moment is

$$M_t = V_1 r = 1{,}500 \times 8 = 12{,}000.$$

The section modulus for torsion

$$\frac{J_t}{a}^* = 0.2\,d^3 = 0.2 \times 5^{-3} = 25.$$

Maximum torsional strain,

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{12{,}000}{25} = 480 \text{ pounds.}$$

The latter strain is small compared with the bending strain, and can be disregarded.

Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

The bending moment is

$$M_b = H_1\,(c_1 - e) = 14{,}300 \times 7 = 100{,}100.$$

Section modulus

$$\frac{J}{a} = \frac{h\,b^2}{6} = \frac{6 \times 9}{6} = 9.$$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{100{,}100}{9} = 11{,}120 \text{ pounds.}$$

The twisting moment in a vertical section of the arm is

$$M_t = V_1\,(c_1 - e) = 1{,}500 \times 7 = 10{,}500.$$

The section modulus

$$\frac{J_t}{a} = \frac{2}{9}\,h\,b^2 = \frac{2}{9}\,6 \times 9 = 12.$$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{10{,}500}{12} = 875 \text{ pounds.}$$

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* Section Modulus for Torsion, $\frac{J_t}{a} = \frac{\text{Polar Mom. of Inertia}}{\frac{d}{2}} = \frac{\pi}{16}\,d^3 =$ appr. $0.2\,d^3$.

The bending and twisting strains are combined into an equivalent tensile strain by equation

$$S = 0.35\, S_b + 0.65 \sqrt{S_b^2 + 4\, S_t^2}.$$

Hence,

$$S = 0.35 \times 11{,}120 + 0.65 \sqrt{\overline{11{,}120}^2 + 4 \times \overline{875}^2} = 11{,}221 \text{ pounds.}$$

The compressive strain in the arm is

$$S_c = \frac{H_1}{b\,h} = \frac{14{,}300}{18} = 794 \text{ pounds.}$$

The total maximum strain in the arm, therefore,

$$S + S_c = 12{,}015 \text{ pounds.}$$

Maximum Strain in the Right-Hand Crank-Arm, at the Face of the Broad Side.—

The bending moment is

$$M_b = H_1\,(c_1 + e) - P\,e = 14{,}300 \times 15 - 28{,}600 \times 4 = 100{,}100.$$

This is the same value as for the left-hand crank-arm as it correctly should be.

The maximum bending strain, therefore,

$$S_b = 11{,}120 \text{ pounds.}$$

The maximum twisting moment is

$$M_t = V_1\,(c_1 + e) = 1{,}500 \times 15 = 22{,}500.$$

The section modulus $\dfrac{J_t}{a} = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\dfrac{J_t}{a}} = \frac{22{,}500}{12} = 1{,}875 \text{ pounds.}$$

The combined strain will be

$$S = 0.35 \times 11{,}120 + 0.65 \sqrt{\overline{11{,}120}^2 + 4 \times \overline{1{,}875}^2} = 11{,}516 \text{ pounds.}$$

The compressive strain in the arm is

$$S_c = 794 \text{ pounds.}$$

The total maximum strain, therefore,

$$S + S_c = 12{,}310 \text{ pounds.}$$

In case there is one wheel on each end of the shaft at the distance f from the bearings, thus $V_1 = V_2 = W$, then the torsional moment in the crank-pin becomes zero, and the bending moment due to H_1, V_1 and W becomes

$$M_b = c_1 \sqrt{H_1^2 + \left(\frac{f}{c_1}\right)^2 V_1^2}.$$

The heaviest strain will occur in the crank-pin and arms when the crank passes the centre. The preceding estimate is therefore sufficient, as far as the strength of the pin and arms are concerned. For comparison, however, a similar estimate for the crank-position 30 degrees from the centre will be carried out in the following:

The Crank 30 Degrees above Centre.—The crank is represented in Fig. 63 in a position 30 degrees from the head-end centre, and the forces acting on the shaft, tangentially and radially to the crank, are as denoted.

The force P_{30} is dissolved into the tangential component T_p and the radial component C_p.

The reactions in the bearings from T_p are T_{p1} and T_{p2}, and the reactions from C_p are C_{p1} and C_{p2}.

The weight of the fly-wheel is W, and T_w and C_w are the tangential and radial components due to the weight W.

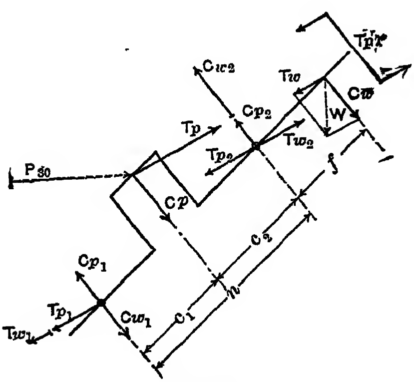


FIG. 63.

The reactions in the bearings from T_w are T_{w1} and T_{w2}, and the reactions from C_w are C_{w1} and C_{w2}.

The turning resistance at the fly-wheel is represented by the moment $T_p . r$, applied at the fly-wheel hub, but the belt-pull, being small, is neglected.

The symbols c_1, c_2 and f denote, according to the figure, as before, the distances between the left-hand and right-hand bearings and the centre of the engine, and between the right-hand bearing and the centre of

the fly-wheel; n is the distance, centre to centre, between the bearings.

The data given are:

$$c_1 = c_2 = 11''.$$
$$f = 11'',$$
$$n = 22''.$$

$$P_{30} = 0.75\,P = 21{,}450 \text{ pounds}$$
$$W = 3{,}000 \text{ pounds.}$$

From these we obtain

$$C_p = \cos(\alpha + \beta)\ P_{30} = 0.817 \times 21{,}450 = 17{,}524$$
$$T_p = \sin(\alpha + \beta)\ P_{30} = 0.576 \times 21{,}450 = 12{,}356$$
$$C_{p1} = \frac{c_2}{n}\,C_p = \tfrac{1}{2} \times 17{,}524 = 8{,}762$$
$$T_{p1} = \frac{c_2}{n}\,T_p = \tfrac{1}{2} \times 12{,}356 = 6{,}178$$
$$C_{p2} = \frac{c_1}{n}\,C_p = \tfrac{1}{2} \times 17{,}524 = 8{,}762$$
$$T_{p2} = \frac{c_1}{n}\,T_p = \tfrac{1}{2} \times 12{,}356 = 6{,}178$$

$$C_w = W \sin 30^\circ = \tfrac{1}{2} \times 3{,}000 = 1{,}500$$
$$T_w = W \cos 30^\circ = 0.866 \times 3{,}000 = 2{,}600$$
$$C_{w1} = \frac{f}{n}\ C_w = \tfrac{1}{2} \times 1{,}500 = 750$$
$$T_{w1} = \frac{f}{n}\ T_w = \tfrac{1}{2} \times 2{,}600 = 1{,}300$$
$$C_{w2} = \frac{n+f}{n}\ C_w = \tfrac{3}{2} \times 1{,}500 = 2{,}250$$
$$T_{w2} = \frac{n+f}{n}\ T_w = \tfrac{3}{2} \times 2{,}600 = 3{,}900$$

$$C_{p1} - C_{w1} = 8{,}012$$
$$T_{p1} + T_{w1} = 7{,}478$$

In Fig. 64 are represented the same forces as are shown in Fig. 63, as well as the sections of the material of the shaft resisting them.

The Maximum Strain at the Middle of the Crank-Pin.—
The moment acting radially

$$M_{b1} = (C_{p1} - C_{w1})\, c_1 = 8{,}012 \times 11 = 88{,}132.$$

The moment acting tangentially

$$M_{b2} = (T_{p1} + T_{w1})\, c_1 = 7{,}478 \times 11 = 82{,}258.$$

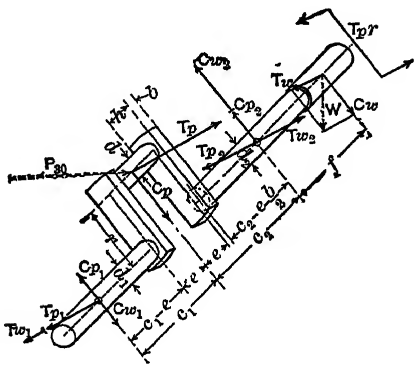


FIG. 64.

The combined moment is

$$M_b = \sqrt{M_{b\,1}^2 + M_{b\,2}^2} = \sqrt{\overline{8{,}012}^2 + \overline{7{,}478}^2} \times 11 = 120{,}000.$$

The section modulus is $\dfrac{J}{a} = 0.1\, d^3 = 12.5.$

The maximum bending strain in pin

$$S_b = \frac{M_b}{\dfrac{J}{a}} = \frac{120{,}000}{12.5} = 9{,}600 \text{ pounds.}$$

The maximum twisting moment is

$$M_t = (T_{p1} + T_{w1})\, r = 7{,}478 \times 8 = 59{,}824.$$

The section modulus for torsion is $\dfrac{J_t}{a} = 0.2\, d^3 = 25.$

The maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{59{,}824}{25} = 2{,}390 \text{ pounds.}$$

The combined strain thus

$$S = 0.35 \times 9{,}600 + 0.65 \sqrt{\overline{9{,}600}^2 + 4 \times \overline{2{,}390}^2} = 10{,}330 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = (C_{p1} - C_{w1})\,(c_1 - e) = 8{,}012 \times 7 = 56{,}084.$$

The section modulus $\frac{J}{a} = \frac{h\,b^2}{6} = \frac{6 \times 9}{6} = 9.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{56{,}084}{9} = 6{,}232 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\,(c_1 - e) = 7{,}478 \times 7 = 52{,}346.$$

Section modulus for torsion $\frac{J_t}{a} = \frac{2}{9}\,h\,b^2 = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{52{,}346}{12} = 4{,}362 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 6{,}232 + 0.65 \sqrt{\overline{6{,}232}^2 + 4 \times \overline{4{,}362}^2} = 9{,}150 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Narrow Side Near Pin.—

Maximum bending moment

$$M_b = (T_{p1} + T_{w1})\left(r - \frac{d}{2}\right) = 7{,}478 \times 5.5 = 41{,}129.$$

The section modulus $\frac{J}{a} = \frac{b\,h^2}{6} = \frac{3 \times 36}{6} = 18.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{41,129}{18} = 2,285 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\ (c_1 - e) = 52,346, \text{ as before.}$$

Section modulus for torsion $\frac{J_t}{a} = \frac{2}{9} b\, h^2 = 24.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{52,346}{24} = 2,181 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 2,285 + 0.65 \sqrt{\overline{2,285}^2 + 4 \times \overline{2,181}^2} = 3,990 \text{ pounds.}$$

The Maximum Strain in the Right-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = (C_{p1} - C_{w1})\ (c_1 + e) - C_p\ e = 8,012 \times 15 - 17,524 \times 4 = 50,084.$$

The section modulus $\frac{J}{a} = 9.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{50,084}{9} = 5,565 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\ (c_1 + e) - T_p\, e =$$
$$7,478 \times 15 - 12,356 \times 4 = 62,746 \text{ pounds.}$$

Section modulus for torsion $\frac{J_t}{a} = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{62,746}{12} = 5,229 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 5{,}565 + 0.65 \sqrt{\overline{5{,}565}^2 + 4 \times \overline{5{,}229}^2} = 9{,}650 \text{ pounds.}$$

The Maximum Strain in the Right-Hand Crank-Arm, at the Face of the Narrow Side Near the Shaft.—

Maximum bending moment

$$M_b = T_p \left(r - \frac{d_2}{2}\right) + (T_{p1} + T_{w1}) \frac{d_2}{2}$$

$$= 12{,}356 + 5.5 + 7{,}478 \times 2.5 = 86{,}653.$$

The section modulus $\dfrac{J}{a} = \dfrac{b\,h^2}{6} = 18.$

Maximum bending strain

$$S_b = \frac{M_b}{\dfrac{J}{a}} = \frac{86{,}653}{18} = 4{,}814 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\,(c_1 + e) - T_p\, e = 62{,}746, \text{ as before.}$$

Section modulus for torsion $\dfrac{J_t}{a} = \dfrac{2}{9}\, b\,h^2 = 24.$

Maximum twisting strain

$$S_t = \frac{M_t}{\dfrac{J_t}{a}} = \frac{62{,}746}{24} = 2{,}614 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 4{,}814 + 0.65 \sqrt{\overline{4{,}814}^2 + 4 \times \overline{2{,}614}^2} = 6{,}304 \text{ pounds.}$$

The Maximum Strain in the Shaft, at the Middle Section of the Main Journal.—When the crank is on centre there is no twisting moment at any section of the main shaft, but there exists, for any position of the crank, a bending moment *at the centre of the journal next to the fly-wheel,* which is,

$$M_b = W\,f = 3{,}000 \times 11 = 33{,}000.$$

The section modulus for the shaft is $\dfrac{J}{a} = 0.1\, d^3_2 = 12.5.$

The maximum bending strain therefore

$$S = \frac{M_b}{\frac{J}{a}} = \frac{33{,}000}{12.5} = 2{,}640 \text{ pounds.}$$

When the crank stands 30 degrees from the head-end centre there is added to this a maximum twisting moment

$$M_t = T_p \times r = 12{,}356 \times 8 = 98{,}848.$$

The section modulus for torsion is $\frac{J_t}{a} = 0.2\, d^3_2 = 25.$

The maximum twisting strain

$$\frac{M_t}{\frac{J_t}{a}} = \frac{98{,}848}{25} = 3{,}954.$$

The combined maximum strain at the centre of the journal

$$S = 0.35 \times 2{,}640 + 0.65\sqrt{2{,}640^{-2} + 4 \times 3{,}954^{-2}} = 6{,}340 \text{ pounds.}$$

The Maximum Strain in the Section of the Main Shaft Next to the Right-Hand Crank-Arm.—

Maximum bending moment radially to the crank

$$M_{b1} = (C_{p2} + C_{w2})\left(c_2 - e - \frac{b}{2}\right) - C_w\left(f + c_2 - e - \frac{b}{2}\right)$$

$$= 11{,}012 \times 5.5 - 1{,}500 \times 16.5 = 85{,}316.$$

Maximum bending moment tangentially to the crank

$$M_{b2} = (T_{p2} - T_{w2})\left(c_2 - e - \frac{b}{2}\right) + T_w\left(f + c_2 - e - \frac{b}{2}\right)$$

$$= 2{,}278 \times 5.5 + 2{,}600 \times 16.5 = 55{,}429.$$

The combined bending moment

$$M_b = \sqrt{M_{b}{}^2_1 + M_{b}{}^2_2} = \sqrt{85{,}316^{-2} + 55{,}429^{-2}} = 101{,}740.$$

The section modulus is $\frac{J}{a} = 0.1\, d^3_2 = 12.5.$

The maximum bending strain

$$\frac{M_b}{\frac{J}{a}} = \frac{101{,}740}{12.5} = 8{,}140 \text{ pounds.}$$

The maximum twisting moment is as before $M_t = T_p\, r = 98{,}848$, and the maximum twisting strain $S_t = 3{,}954$ pounds.

The combined maximum strain will be

$$S = 0.35 \times 8{,}140 + 0.65 \sqrt{\overline{8{,}140}^2 + 4 \times \overline{3{,}954}^2} = 10{,}220 \text{ pounds.}$$

The Strength of a Centre-Crank Shaft Supported on Three Bearings.—In Fig. 65 are represented all the forces acting on a shaft of this type, when the crank, at the time of the explosion, passes the centre.

P is the total pressure on the piston, H_1 and H_2 the reactions on the bearings 1 and 2 due to this force. *W* is the weight of the fly-wheel, and the reactions due to it in the bearings 2 and 3 are V_2 and V_3.

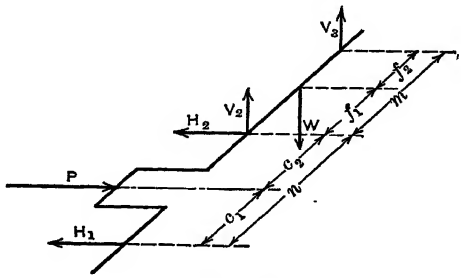


FIG. 65.

Assume it to be required to analyze the strains in a shaft for a 20 × 32 engine, the preliminary sketch of which is shown in Fig. 66.

Running on suction producer-gas at 160 revolutions per

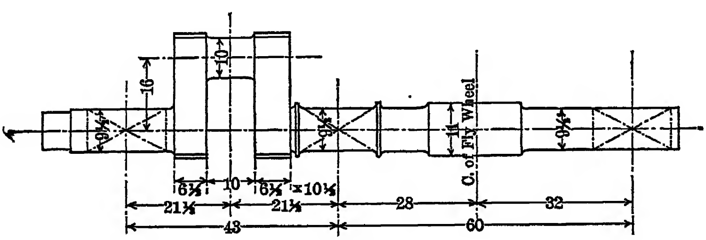


FIG. 66.

minute, the power of the engine will be approximately 110 B.H.P., or 144 I.H.P. A heavy wheel would be of a weight of 32,000 pounds, of a diameter 11′ — 0″.

The area of the piston being 314 square inches, and the maximum pressure allowed at 450 pounds per square inch, we obtain

the total maximum pressure on the piston $P = 314 \times 450 = 141{,}300$ pounds.

The maximum pressure on the crank when 30 degrees from the head-end centre will be $P_{30} = 0.75\ P = 106{,}000$ pounds.

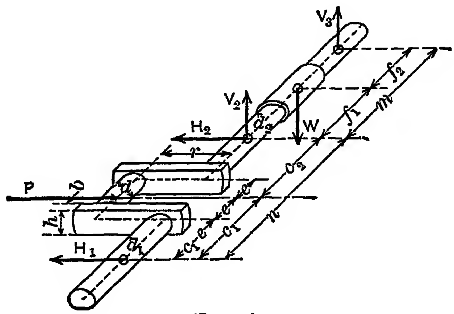


FIG. 67.

The principal dimensions of the shaft as well as the forces acting, when the crank passes the centre, are given in Fig. 67.

The data given are:

$$\begin{array}{lll} c_1 = c_2 = 21.5'', & e = 8.25'' & \\ c_1 - e = 13.25'', & f_1 = 28'', & f_2 = 32''. \\ n = 43'', & b = 6\tfrac{1}{2}'', & d = 10'', \\ m = 60'', & h = 10\tfrac{1}{2}'', & r = 16''. \end{array}$$

$$P = 141{,}300 \text{ pounds.}$$
$$W = 32{,}000 \text{ pounds.}$$

From this we obtain

$$H_1 = H_2 = \frac{c_2}{n}\ P = 70{,}650 \text{ pounds}$$

$$V_2 = \frac{f_2}{m} W = \tfrac{32}{60} \times 32{,}000 = 17{,}067 \text{ pounds.}$$

$$V_3 = \frac{f_1}{m} W = \tfrac{28}{60} \times 32{,}000 = 14{,}933 \text{ pounds.}$$

The Maximum Strain at the Middle of the Crank-Pin.
Maximum bending moment

$$M_b = H_1\ c_1 = 70{,}650 \times 21.5 = 1{,}518{,}975.$$

The section modulus is $\dfrac{J}{a} = 0.1\ d^3 = 0.1 \times 10^{-3} = 100.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = 15{,}190 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = H_1\,(c_1 - e) = 70{,}650 \times 13.25 = 936{,}112.$$

The section modulus is $\frac{J}{a} = \frac{h\,b^2}{6} = \frac{10.5 \times 6.5^{-2}}{6} = 74.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{936{,}112}{74} = 12{,}650 \text{ pounds.}$$

The strain due to compression

$$S_c = \frac{H_1}{b \times h} = \frac{70{,}650}{68.25} = 1{,}035 \text{ pounds.}$$

The total maximum strain in arm $S = 13{,}685$ pounds.

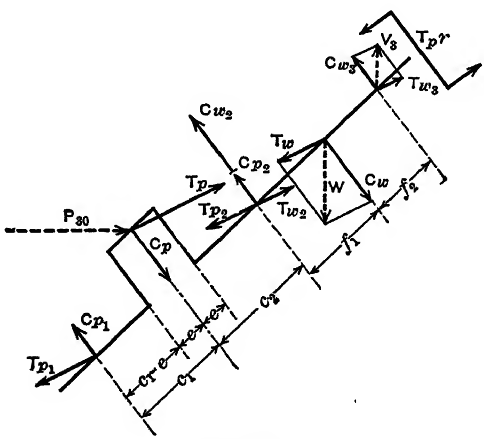


FIG. 68.

The Maximum Strains in the Right-Hand Crank-Arm, at the Face of the Broad Side are the same as above.

The strains in the shaft, at the fly-wheel hub, at the middle

of the main bearing, and in a section near the right-hand arm, should be figured with respect to the forces acting on the shaft when the crank stands 30 degrees from the head-end centre. These forces are shown diagrammatically by Fig. 68. P_{30} is the force transmitted by the connecting-rod to the crank-pin and C_p and T_p its components, radially and tangentially to the crank.

The component C_p gives the reactions C_{p1} and C_{p2} at the bearings 1 and 2 and the component T_p gives the reactions T_{p1} and T_{p2}. The fly-wheel weight is represented by W, and its components, radially and tangentially, are C_w and T_w.

The reactions in bearings 2 and 3, due to the radial component,

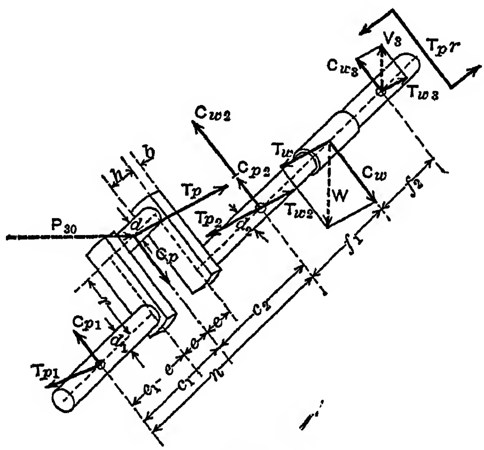


FIG. 69.

C_w, are respectively C_{w2} and C_{w3}, and the reactions due to the tangential component, T_w, are T_{w2} and T_{w3}. $T_p.r$ is a turning moment acting at the fly-wheel hub, representing the resistance to the rotation.

In Fig. 69 are represented the same forces as are shown in Fig. 68, as well as the sections of the material resisting them.

The data given are:

$$c_1 = c_2 = 21.5'', \qquad f_1 = 28'', \qquad f_2 = 32,''$$
$$m = 60'', \qquad r = 16'', \qquad e = 8.25'',$$
$$d_1 = d_2 = d_3 = 9\tfrac{1}{2}'', \qquad d_4 = 11''.$$

$$P_{30} = 0.75\ P = 106{,}000 \text{ pounds.}$$
$$W = 32{,}000 \text{ pounds.}$$

From this is obtained

$$C_p = cos\ (\alpha + \beta)\ P_{30} = 0.817 \times 106{,}000 = 86{,}600 \text{ pounds}$$
$$T_p = sin\ (\alpha + \beta)\ P_{30} = 0.576 \times 106{,}000 = 61{,}056 \text{ pounds}$$
$$C_{p2} = \frac{c_1}{n}\ C_p = \tfrac{1}{2}\ 86{,}600 = 43{,}300 \text{ pounds.}$$
$$T_{p2} = \frac{c_1}{n}\ T_p = \tfrac{1}{2}\ 61{,}056 = 30{,}528 \text{ pounds.}$$

$$C_w = W\ sin\ 30° = \tfrac{1}{2}\ 32{,}000 = 16{,}000 \text{ pounds.}$$
$$T_w = W\ cos\ 30° = 0.866 \times 32{,}000 = 27{,}712 \text{ pounds.}$$
$$C_{w2} = \frac{f_2}{m}\ C_w = \tfrac{32}{60}\ 16{,}000 = 8{,}533 \text{ pounds.}$$
$$T_{w2} = \frac{f_2}{m}\ T_w = \tfrac{32}{60}\ 27{,}712 = 14{,}780 \text{ pounds.}$$
$$C_{w3} = \frac{f_1}{m}\ C_w = \tfrac{28}{60}\ 16{,}000 = 7{,}466 \text{ pounds.}$$
$$T_{w3} = \frac{f_1}{m}\ T_w = \tfrac{28}{60}\ 27{,}712 = 12{,}930 \text{ pounds.}$$

$$C_{p2} + C_{w2} = 51{,}833 \text{ pounds.}$$
$$T_{p2} - T_{w2} = 15{,}748 \text{ pounds.}$$

The resultant of the forces C_{w3} and T_{w3} we obtain as

$$V_3 = \frac{f_1}{m}\ W = \tfrac{28}{60}\ 32{,}000 = 14{,}933.$$

The Maximum Strain at the Centre of the Fly-Wheel. Maximum bending moment

$$M_b = V_3 \times f_2 = 14{,}933 \times 32 = 477{,}856.$$

The section modulus $\dfrac{J}{a} = 0.1\ d^3_4 = 0.1 \times 11^3 = 133.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{477{,}856}{133} = 3{,}590 \text{ pounds.}$$

The Maximum Strain at the Middle of Journal 2.—
Maximum bending moment

$$M_b = V_3\, m - W\, f_1 = 0.$$

Maximum twisting moment

$$M_t = T_p\, r = 61{,}056 \times 16 = 976{,}896.$$

The section modulus for torsion

$$\frac{J_t}{a} = 0.2\, d^3_2 = 0.2 \times \overline{9.5}^3 = 171.$$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{976{,}896}{171} = 5{,}713 \text{ pounds.}$$

Maximum Strain at the Section of the Shaft next to the Right-Hand Crank-Arm.—

The bending moment at bearing 2 being zero; we have, for the section next to the right-hand crank-arm,

Maximum bending moment radially to the crank

$$M_{b1} = (C_{p2} + C_{w2}) \left(c_2 - e - \frac{b}{2}\right) = 51{,}833 \times 10 = 518{,}330.$$

Maximum bending moment tangentially to the crank

$$M_{b2} = (T_{p2} - T_{w2}) \left(c_2 - e - \frac{b}{2}\right) = 15{,}748 \times 10 = 157{,}480.$$

The combined bending moment becomes

$$M_b = \sqrt{M_{b1}^2 + M_{b2}^2} = \sqrt{\overline{518{,}330}^2 + \overline{157{,}480}^2} = 542{,}000.$$

The section modulus is $\frac{J}{a} = 0.1\, d^3_2 = 85.7.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{542{,}000}{85.7} = 6{,}320 \text{ pounds.}$$

The twisting moment is $M_t = T_p\, r = 976{,}896.$

The section modulus for torsion is $\frac{J_t}{a} = 171.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J}{a}} = \frac{976,896}{171} = 5,712 \text{ pounds.}$$

The combined maximum strain becomes

$$S = 0.35 \times 6,320 + 0.65 \sqrt{6,320^2 + 4 \times 5,712^2} = 10,700 \text{ pounds.}$$

Shaft with Side-Crank.—Let it be assumed that Fig. 70 is a proposed shaft for a 20 × 32 blast-furnace gas-engine.

At the time of the combustion in the cylinder, when the crank

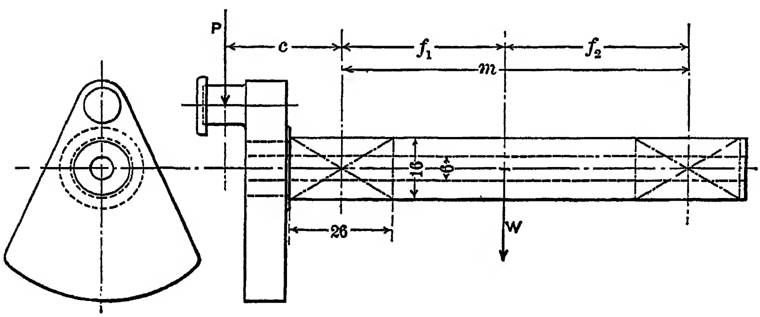


FIG. 70.

passes the centre, the full pressure, P, on the piston will be transmitted to the crank-pin, and the pressure on the main bearing will be $\frac{m+c}{m}P$. The approximate bearing surface of the journal is determined according to equation 110 to suit this pressure and it is found that a projected bearing surface 16″ × 26″ will be ample. In order to obtain a good connection between the crank and shaft it will be required to make the distance c = 28 inches. For a maximum pressure 450 pounds per square inch of the piston, the total pressure on the crank-pin will be P = 141,000 pounds.

According to these data, the strength calculation for the shaft will be as follows:

The Maximum Strain in the Shaft at the Middle Section of the Main Journal.—

Maximum bending moment

$$M_b = P \times c = 141{,}000 \times 28 = 3{,}948{,}000.$$

The section modulus for a hollow shaft

$$\frac{J}{a} = 0.1\,\frac{d^4 - d^4_1}{d} = 0.1\,\frac{16^4 - 6^4}{16} = 400.$$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{3{,}948{,}000}{400} = 9{,}870 \text{ pounds.}$$

The section modulus for a solid shaft will be

$$\frac{J}{a} = 0.1\,d^3 = 410.$$

and the maximum bending strain

$$S_b = \frac{3{,}948{,}000}{410} = 9{,}630 \text{ pounds.}$$

When the crank stands 30 degrees past the centre, as Fig. 71, we have $P_{30} = 0.75\,P = 106{,}000$ pounds.

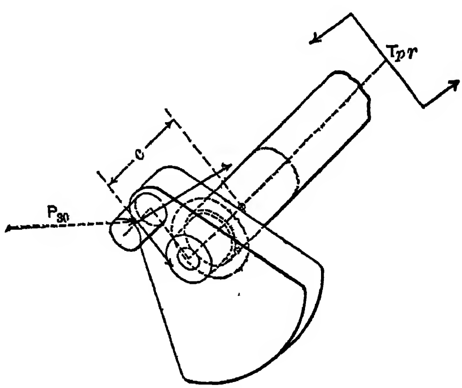


FIG. 71.

The Maximum Strain at the Middle Section of the Journal. Crank 30° Above Centre.—

Maximum bending moment,

$$M_b = P_{30} \times c = 106{,}000 \times 28 = 2{,}968{,}000.$$

Section modulus for a hollow shaft $\frac{J}{a} = 400.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{2{,}968{,}000}{400} = 7{,}420 \text{ pounds.}$$

Maximum torsional moment

$$M_t = P_{30}\, r \sin(\alpha + \beta) = 106{,}000 \times 16 \times 0.576 = 976{,}900.$$

The section modulus for torsion $\frac{J_t}{a} = 2 \times \frac{J}{a} = 800.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{976{,}900}{800} = 1{,}220 \text{ pounds.}$$

The combined maximum strain is

$$S = 0.35 \times 7{,}420 + 0.65 \sqrt{7{,}420^{-2} + 4 \times 1{,}220^{-2}} = 7{,}670 \text{ pounds.}$$

Deflection of the Shaft.—When a heavy wheel or generator armature is carried on a long shaft, between the main and outboard journals, it will be necessary to analyze the shaft as to its stiffness.

The deflection of the shaft, expressed in inches, is given by the formula

$$\delta = \tfrac{1}{3} \frac{P}{J\,E} \frac{f_1^2 \times f_2^2}{m} \text{ inches.} \quad . \quad . \quad . \quad (109)$$

P is the total load carried,
J the moment of inertia of the cross-section $= 0.05\ d^4$,
E the coefficient of elasticity, for steel 30,000,000,
m the distance between the journals, in inches,
f_1 and f_2 the distances from the centre of the wheel, respectively, to the main- and the outboard-bearing.

When the load is applied approximately central between the bearings the equation becomes

$$\delta = \frac{1}{48} \frac{P}{J\,E} m^3. \quad . \quad . \quad . \quad . \quad . \quad (109a)$$

Assume that a 10-inch shaft, carrying a 32,000-pound wheel,

as shown in Fig. 70, is supported by bearings with 80-inch centres. The weight of the shaft is 2,000 pounds, which added to the weight of the wheel gives the total load 34,000 pounds located centrally between the bearings. The weight of the shaft being small, compared with the weight of the wheel, it may simply be added to the latter weight.

According to the data given, we obtain

$$\delta = \frac{1}{48} \frac{34,000 \times 80^3}{0.05 \times 10^4 \times 30,000,000} = 0.024 \text{ inch.}$$

The deflection of a shaft carrying an electric generator-armature should generally not be allowed to exceed 0.03 inch, and a proper allowance should be made for the magnetic pull on the armature, when below the true centre.

Allowable Strains in the Shaft due to the unreduced maximum pressure on the piston.

The weakest sections of a centre-crank shaft, as generally carried out, are the crank-pin, and the section of the shaft near the crank-arm toward the wheel. In order to avoid the necessity of an excessively heavy connecting-rod, the crank-pin is made small, within safe limits. A maximum strain of 12,000 to 15,000 pounds in the material of the pin can safely be allowed.

The highest strain in the shaft proper should not, however, be more than 11,000 to 13,000 pounds.

There being generally no good reason for making the crank-arms very light, and as the material put into them contributes essentially to the rigidity of the complete shaft, a strain in the arms of 11,000 to 13,000 pounds per square inch is generally not exceeded.

These strains in the shaft may appear heavy, but it will be understood that they do not represent the average maximum working strains but the maximum strains to which the shaft may occasionally be subjected.

CHAPTER XI

ENGINE DETAILS

The Engine-Bed.—Fig. 72 illustrates a type of engine-bed that has been adopted by several builders, for engines of sizes up to 22 inches cylinder-diameter. It is originally of German design and is adapted for a centre-crank shaft.

There are, however, as has been pointed out, some serious objections to the employment of centre-cranks in connection with large twin engines, on account of the necessity of installing and maintaining four or more bearings in alignment for the proper support of the shaft, and on account of the couplings that frequently will be required for coupling the two engine shafts together. The type of engine-bed illustrated in Fig. 73, requiring in any case not more than two journals, is to be preferred for large engines, assuming that there is no serious objection to the expense of a heavy shaft.

The main requirement in an engine-bed is rigidity with respect to the heavy strains that act between the journals and the rear connection to the cylinder. This requirement is, in the engine-beds illustrated, very well provided for, by the employment of deep girder-designs that offer adequate stiffness against the bending forces.

Strain in the Bed.—Fig. 74 represents the journal-end of a side-crank engine-frame. At the crank-pin there is applied a maximum force $P = A \times 450$, which reduced to the centre of the journal becomes P_1.

Let it be desired to find the maximum strain in the section $A\ B$ of an engine-frame for a 24-inch cylinder of the type illustrated, and of the detail-dimensions as given in connection with Fig. 74, on page 255.

The forces acting are:

$P\ = 452 \times 450 = 203{,}400$ pounds;
$P_1 = \qquad\qquad\quad 220{,}000$ pounds;

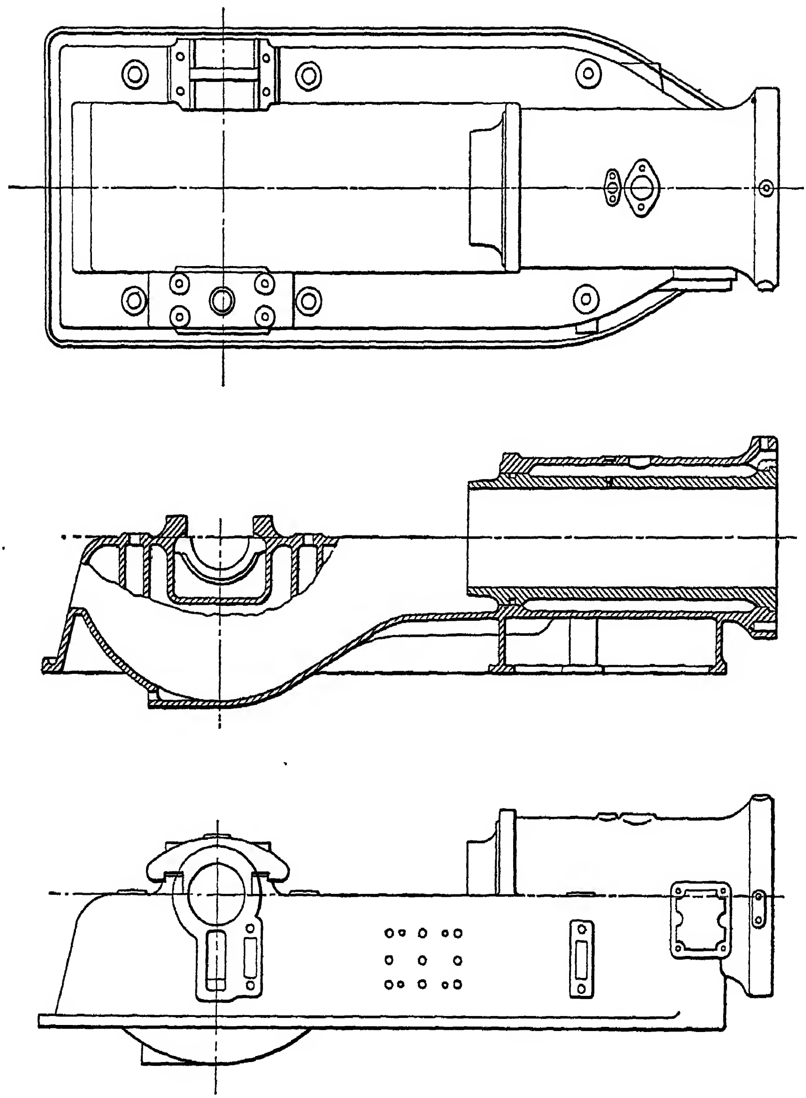


FIG. 72.—Engine-Bed for Centre-Crank Shaft.

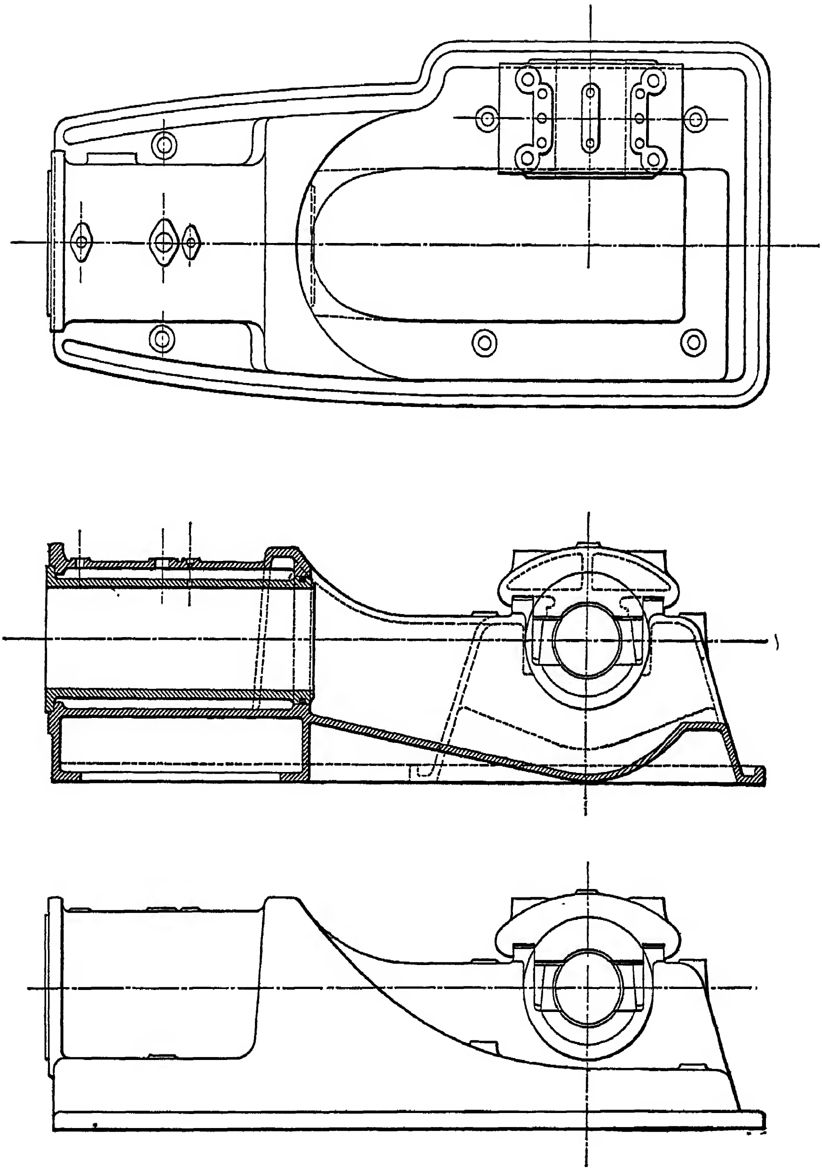


FIG. 73.—Engine-Bed for Side-Crank Shaft.

The bending moment in section $A\ B$ is

$$M_b = P_1 \times l = 220{,}000 \times 11 = 2{,}420{,}000.$$

With reference to the neutral axis $C\ D$:

Moment of inertia of two side walls

$$J_1 = \frac{2\, t_2\, h^3}{12} = \frac{2 \times 1.5 \times 42^{-3}}{12} = 18{,}522.$$

Moment of inertia of top side

$$J_2 = \frac{b\, t_1^3}{12} + t_1\, b\, a_1^2 = \frac{12 \times 1.75^{-3}}{12} + 1.75 \times 12 \times 20.12^{-2} = 8{,}520.$$

Moment of inertia of bottom flanges

$$J_3 = \frac{(c - d - 2t_2)\, t^3_3}{12} + (c - d - 2\, t_2)\, t_3\, a_3^2$$

$$= \frac{(21 - 4 - 3)\, 1.5^{-3}}{12} + (21 - 4 - 3)\, 1.5 \times 20.25^{-2} = 8{,}614.$$

The section modulus of the whole section

$$\frac{J}{a} = \frac{J_1 + J_2 + J_3}{a} = \frac{35{,}656}{21} = 1{,}700.$$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{2{,}420{,}000}{1{,}700} = 1{,}420 \text{ pounds.}$$

The tensile strain is =

$$\frac{P_1}{\text{area of section}} = \frac{220{,}000}{168} = 1{,}310 \text{ pounds.}$$

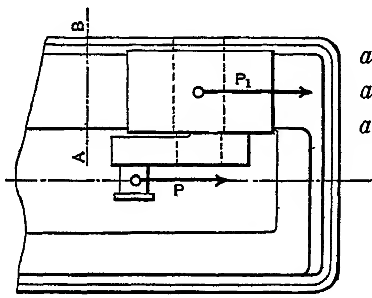
Total maximum strain in section $A\ B$,

$$S = 1{,}310 + 1{,}420 = 2{,}730 \text{ pounds.}$$

The Crank-Pin and Piston-Pin Journals.—The maximum pressure per square inch projected surface of the journals, due to the maximum pressure on the piston, can in a gas-engine be allowed higher than what is generally the practice with respect to the steam-engine. This is justified for the reason that the maximum pressure on the gas-engine piston is of short duration

only; hence, the heat evolved by friction is less than in the steam-engine, and an effective lubrication of the journals is more readily maintained. The piston-pin is often made small as a matter of necessity, on account of limited room, but it is with advantage kept as large as circumstances will allow.

The service for which an engine is intended and the grade of the material put into its pins and journal-boxes will, of course, greatly influence their minimum safe dimensions. In practice,



For a 24-inch engine:

$a = 21''$, $b = 12''$,
$a_1 = 20\frac{1}{8}''$, $c = 21''$,
$a_3 = 20\frac{1}{4}''$, $d = 4''$,
$t_1 = 1\frac{3}{4}''$,
$t_2 = 1\frac{1}{2}''$,
$t_3 = 1\frac{1}{2}''$,
$h = 42''$,
$l = 11''$.

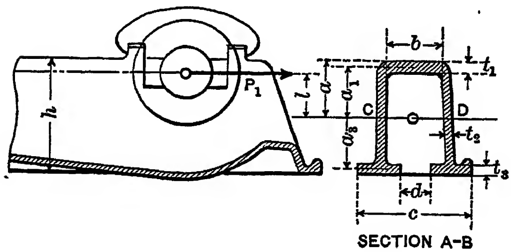


FIG. 74.

therefore, engines required to be as light as possible, and built with particular care, have often pins and bearing-surfaces proportionally much smaller than what would be safe to use in heavier stationary engines. The piston-pin and crank-pin of small engines can, with equal safety, also be made to bear a higher breaking strain, as well as a higher bearing-pressure per square inch projected surface, than engines of medium and high power.

Hence, when considering, in a general way, the limiting size of the wearing surfaces in engines of different construction, distinction should be made, in the first place, between small engines,

and engines of medium and large power. Secondly, between light-built and heavy engines.

The maximum pressures per square inch of projected surface of the journals safely allowed in practice are:

TABLE XXIII.

	On the Main Bearing.		On the Crank Pin.		On the Piston Pin.	
	Heavy Engines.	Light Engines.	Heavy Engines	Light Engines.	Heavy Engines.	Light Engines.
			B_c Max.		B_w Max.	
Small Engines	400	550	1800	2400	2400	3400
Medium and Large Engines	400	550	1300	1800	1800	2400

The higher figures of the table refer to light-built engines and the lower figures to relatively heavy ones.

The actual maximum pressure on the journals, when the engine is up to speed, is modified very considerably by the inertia of the reciprocating parts, but as a guide for determining the size of the pins the maximum pressure evolved at slow speed is most conveniently used, and this pressure is the basis for the figures in the table above.

It is to be noted, that the maximum pressure on the piston varies in different engines. In engines of high compression of the charge, such as producer-gas or blast-furnace gas engines, the maximum pressure should, with respect to strength and maximum bearing-pressure on the pins, be figured at 450 pounds per square inch, whereas in ordinary, low-compression, gas- or gasoline engines 350 pounds per square inch may be ample. On this account it is necessary to distinguish between the following four classes of engines:

Small engines (below 11 inches cylinder diameter) in which $P_{max.}$ = 350 pounds;

Small engines in which $P_{max.}$ = 450 pounds;

Medium and large engines in which $P_{max.}$ = 350 pounds;

Medium and large engines in which $P_{max.}$ = 450 pounds.

A reduction for the inertia of the reciprocating parts can, if desired, readily be made in these pressures, when the weight of the reciprocating parts is known.

To overcome the inertia of the weight G there is required, at the beginning of the stroke, a total pressure on the piston:

$$P_1 = 0.000034\, G\, N^2\, r.$$

With respect to the piston-pin pressure of a trunk-piston engine, G is the weight of the piston and piston-pin, and with respect to the crank-pin pressure; G is the weight of the piston, piston-pin and one-half of the connecting-rod; N is the number of revolutions per minute, and r is the crank-radius, in inches.

The coefficient 0.000034 is, as stated on page 195, only approximate, but for practical purposes it answers very well.

If the force P_1 be subtracted from the total initial pressure on the piston the remainder will be the actual maximum pressure transmitted to the pin at normal speed.

Constructive considerations limit the length of the trunk-piston-pin, l_w, to approximately one-half the cylinder-diameter.

Thus, $$l_w = 0.5\ D.$$

The diameter of the pin, d_w, may also conveniently be expressed as a percentage of the cylinder-diameter, as

$$d_w = x_w\, D.$$

Hence, $$l_w\, d_w = 0.5\, x_w\, D^2,$$

and, when $B_{w\ max.}$ is the maximum bearing-pressure allowed per square inch projected surface of the wrist-pin, we have

$$l_w\, d_w\, B_{w\ max.} = \frac{\pi\, D^2}{4} P_{max.}$$

and $$x_w = \frac{\pi}{2}\, \frac{P_{max.}}{B_{w\ max.}}$$

At a preliminary estimate of the crank-pin for bearing-pressure it will be convenient to assume its length, l_c, to be the same as that of the wrist-pin,

thus $$l_c = 0.5\ D;$$

and if, similarly as before, the pin-diameter, d_c, be expressed in terms of the cylinder-diameter, or

$$d_c = x_c D,$$

we get

$$x_c = \frac{\pi}{2} \frac{P_{max.}}{B_{c\,max.}}.$$

The values of x_w and x_c solved from the above equations for the limiting values $B_{w\,max.}$ and $B_{c\,max.}$ of Table XXIII, with respect to light and heavy engines, and for $P_{max.}$ 350 and 450 pounds per square inch, will be found in the following table:

TABLE XXIV.

Limiting Sizes of Pins for Different Classes of Engines.

Piston Pin, $d_w = x_w D$, $l_w = 0.5\ D$.

	Small Engines.			
B_w Max.	Light. 3400.		Heavy. 2400.	
P Max. = 350...	$d_w = 0.16\ D$	$S_w = 42700$	$d_w = 0.23\ D$	$S_w = 14000$
P Max. = 450...	$d_w = 0.21\ D$	$S_w = 24400$	$d_w = 0.3\ D$	$S_w = 8330$

	Large Engines.			
B_w Max.	Light. 2400.		Heavy. 1800.	
P Max. = 350...	$d_w = 0.23\ D$	$S_w = 14000$	$d_w = 0.3\ D$	$S_w = 6500$
P Max. = 450...	$d_w = 0.3\ D$	$S_w = 8330$	$d_w = 0.39\ D$	$S_w = 3800$

Crank Pin, $d_c = x_c D$, $l_c = 0.5\ D$

	Small Engines.		Large Engines.	
B_c Max.	Light. 2400.	Heavy. 1800.	Light. 1800.	Heavy. 1300.
P Max. = 350...	$d_c = 0.23\ D$	$d_c = 0.3\ D$	$d_c = 0.3\ D$	$d_c = 0.42\ D$
P Max. = 450...	$d_c = 0.29\ D$	$d_c = 0.39\ D$	$d_c = 0.39\ D$	$d_c = 0.54\ D$

An approximate estimate of the fibre-stress in the wrist-pin may be obtained by figuring its strength as if it were a circular beam uniformly loaded with the total pressure $\frac{\pi D^2}{4} P_{max.}$, and supported at the ends, immediately outside the journal.

The equation for the safe load on the pin will, accordingly, be:

$$\frac{\pi D^2}{4} P_{max.} = 8 \frac{S_{w\ max.}}{l_w} \frac{J}{a};$$

$S_{w\ max.}$ being the allowable fibre-stress,

$\frac{J}{a}$ the section modulus, which is $= \frac{\pi}{32} d_w^3 = \frac{\pi}{32} x_w^3 D^3$,

and $$l_w = 0.5\, D;$$

whence $$S_{w\ max.} = \frac{1}{2\, x_w^3} P_{max.}$$

By solving $S_{w\ max.}$ for $P_{max.} = 350$ and 450 and for the limiting values of x_w of Table XXIV, the approximate fibre-stress in the corresponding pins may be obtained. These stresses, S_w, are given in the table.

The data regarding the bearing pressure allowed, and the dimensions of the crank-pin, alone, are insufficient for determining the fibre-stress in the pin. This stress cannot, therefore, be given in Table XXIV.

The shearing stress in the two sections of the wrist-pin next to where it is supported in the piston will be obtained by multiplying the strain S_w by the factor x_w.

The shearing stress in the end-sections of a wrist-pin of the dimensions: $d_w = 0.16\ D$, and $l_w = 0.5\ D$, when $P_{max.} = 350$ pounds becomes $0.16 \times 42{,}700 = 6{,}832$ pounds per square inch.

Table XXIV will be of convenience as a guide for determining suitable pin-sizes for an engine. Assuming the pressure on the piston to be, for instance, 350 pounds per square inch, then a wrist-pin of a diameter of 0.16 D, which is the smallest pin used in practice, will carry a bearing-pressure 3,400 pounds per square inch projected surface (neglecting the influence due to acceleration at speed) and the fibre-stress in such a pin would be approximately 42,700 pounds. Under the same conditions, a wrist-pin of a diameter 0.3 D — a large pin, will carry a bearing-pressure of 1,800 pounds per square inch, and the approximate fibre-stress will be 6,500 pounds. Similarly, a crank-pin of a diameter 0.23 D will carry a bearing-pressure of 2,400 pounds, and a crank-pin of a diameter 0.42 D a pressure of 1,300 pounds per square inch projected surface.

The intensity of the bearing-pressure on the wrist-pin being occasionally very high it is necessary that the pin be made of a hard material, or, when made of mild steel, it must be case-hardened.

A piston-pin of a diameter of 0.16 of the cylinder-diameter and of a length of 0.5 the cylinder-diameter, carrying a working stress of 42,700 pounds, figured due to the bending of the pin, and a stress 6,832 pounds, figured for shear at the ends, would of course be considered altogether too light. Such pin-diameters are, however, sometimes found in practice, but the length must be made smaller than ½ D; the bearing-pressure, thus, allowed higher than 3,400 pounds as assumed in the table. The length of the pin is rarely actually made over 1¾ to 2 times its diameter.

Main Shaft Journals.—In determining the size of the main journals of a gas-engine, it is convenient to consider the load on the bearings as resulting in two different ways:

First, the load due to the maximum pressure on the piston may be assumed to be transported directly to the bearing, without any reduction for inertia forces, in which case the maximum intensity of the pressure per square inch projected surface becomes $\frac{P_{max} A}{l d}$. As this force acts only during a short time, under the best conditions for free lubrication, and as it actually, under normal speed, will be considerably reduced due to inertia forces, it can be allowed as high as between 275 to 550 pounds per square inch.

$$\text{Thus, generally, } \frac{P_{max.} A}{l d} < 550 \text{ pounds.} \qquad (110)$$

Secondly, the total pressure on the journal may be considered as made up, partly, of the forces transmitted to the shaft from the variable pressure on the piston during the entire cycle, and, partly, of the constant vertical pressure which is due to the weight of the shaft, fly-wheel or generator-armature.

The mean intensity, P_h, of the forces acting on one side or the other of the journal, due to the pressure on the piston, will, of course, vary with the number, and arrangement, of the cylinders. It will, on an average, be

$$P_h = \frac{F A}{2 l d} \qquad (111)$$

The coefficient F having the following values:

in a four-cycle, single-acting single-cylinder engine	$F = 40$
in a four-cycle, single-acting two-cylinder opposed engine	$F = 60$
in a four-cycle, single-acting two-cylinder tandem engine	$F = 70$
in a four-cycle, double-acting one-cylinder engine	$F = 60$
in a four-cycle, double-acting two-cylinder tandem engine	$F = 100$
in a two-cycle single-cylinder engine	$F = 110$

A is the area of one cylinder of one engine, in square inches; l is the length of one bearing, and d its diameter, in inches. A twin engine is only a duplication of a single engine, and the average pressure in the bearing will be the same as in the single engine of the same type.

For a two-cylinder engine with three bearings, or a three-cylinder engine with four bearings, l will be the length of the journal which takes the greatest load due to the weight of the wheels, etc., generally the outside journal.

If V is the pressure on the journal due to the weight of shaft, wheels, etc., the mean pressure per square inch of projected surface of the journal becomes $P_v = \frac{V}{l\,d}$ . . . . . (112)

The heating of the journal is caused by the pressures P_h and P_v, and it will depend also on the surface-speed between the shaft and the bearing.

For cool running, with ordinary lubrication, it is required that

$$P_h + P_v = \frac{\frac{1}{2} F A + V}{d\,l} < \frac{700}{\sqrt{d \frac{N}{60}}}; \quad . \quad . \quad (113)$$

d being the diameter of the journal in inches, and

$\frac{N}{60}$ the number of turns per second.

Main Bearings.—The main bearings of engines up to twenty inches cylinder-diameter are generally provided with babbitt-lined, removable shells, divided on the horizontal line. The construction of an ordinary bearing of this type is shown in Fig. 75. For larger engines adjustable quarter-boxes, as shown in Fig. 73, would be preferable. One quarter-box may be sufficient, par-

ticularly in single-acting engines, and it is properly placed on the side toward the cylinder, opposite the side on which the principal pressure acts.

For the outboard bearing, which carries only a vertical load,

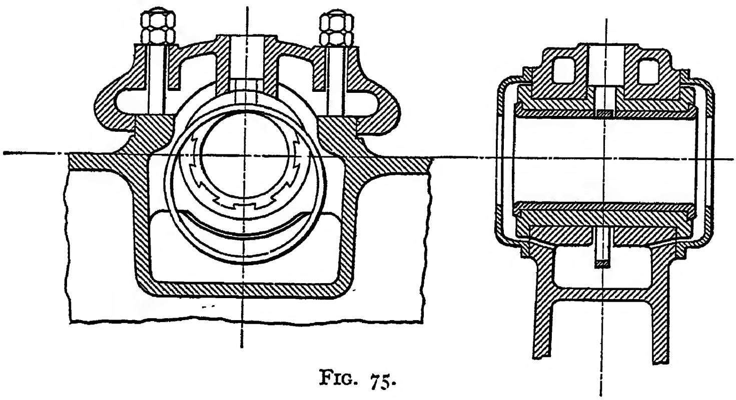


FIG. 75.

no loose boxes are, as a rule, provided. This bearing is, however, often placed on a sole-plate provided with adjusting screws to facilitate the horizontal adjustment of the bearing, at erection, or whenever adjustment would become necessary.

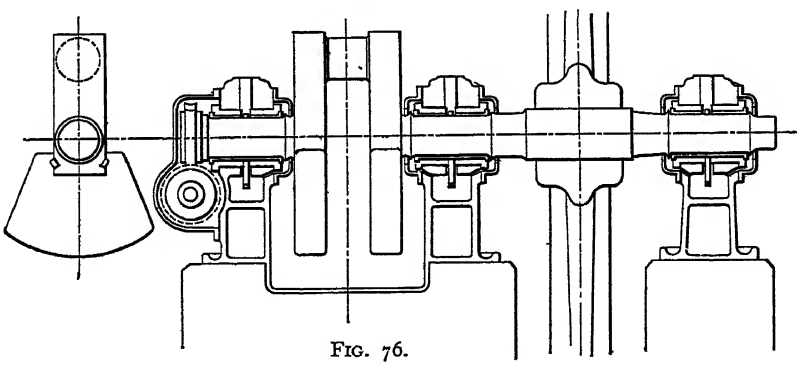


FIG. 76.

Lubrication.—The ring-oiling system of lubrication is simple and effective, but great care should be taken to design the ring and channel for the same so that the ring will run perfectly free, and so that there will be no liability for it to stick. Account

should be taken of the fact that the ring swings out of the plumb-line, when revolving.

For long journals two rings are generally used, placed on a distance of one-third of the length of the journal apart. The rings are made in halves, hinged together, and the open joint dovetailed together, and secured in some way so that it cannot, on any account, open up and get stalled. Large rings should be made of some hard material, that is not readily bent out of shape when the ring is put in place.

Fig. 76 shows an arrangement for the oiling of the main journals, in detail.

The principal trouble experienced with ring-oiling bearings

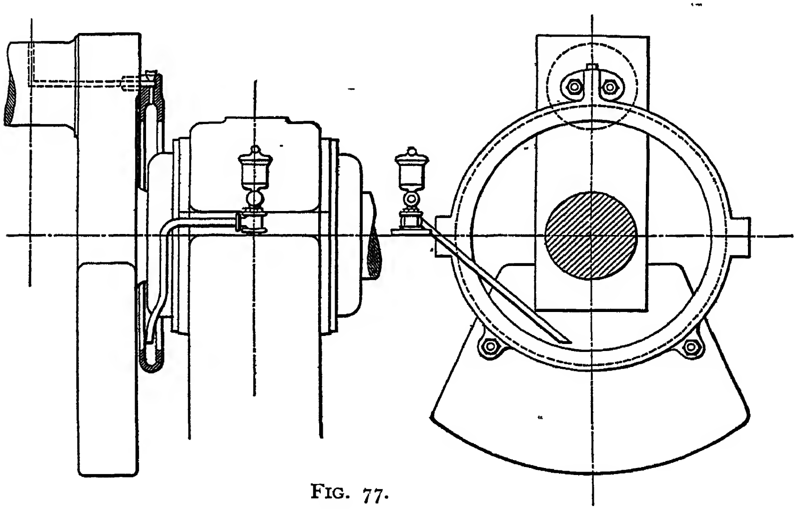


FIG. 77.

has been due to the unexpected stalling of the rings, from one cause or other. In a large plant, a continuous automatic oiling system is, of course, the most modern and most reliable.

The oiling of the crank-pin of a centre-crank is often effected as shown in Fig. 77. The arrangement consists of a cup-ring surrounding the shaft, which catches the oil delivered by the sight-feed cup, and carries it by the action of the centrifugal force through the oil pipe to the crank-pin.

A common oiling arrangement for the piston-pin is illustrated

in Fig. 78. The oil is caught by the extended wiper-cup and carried back to the oil-receptacle formed in the top of the rod-end.

The Piston.—The highest normal pressure between the piston and the cylinder, due to the reaction from the connecting-rod, occurs when the crank stands at an elevation of approximately

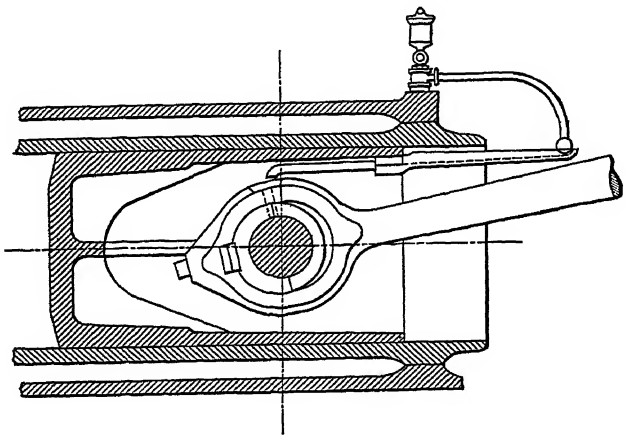


FIG. 78.

30 degrees from the head-end centre. This pressure is, in Fig. 79, designated N.

For the ratio $\frac{r}{l} = \frac{1}{5.5}$, and $\alpha = 30°$, we have

$$\beta = 5° - 10' \quad \text{and} \quad \tan \beta = 0.09.$$

As shown, page 228,

$$P_{30} = 0.75\, P, \text{ approximately,}$$

thus $N_{max.} = 0.09\, P_{30} = 0.09 \times 0.75\ P = 0.068\, P.$

The effective bearing surface of the piston may be counted as being

$$F = 0.85\, D \times 0.8\ L = 0.68\, D\, L.$$

Hence, the bearing pressure per square inch of effective piston-surface

$$p_n = \frac{0.068\ P}{0.68\ D\, L} = \frac{P}{10\, D\, L}.$$

This pressure, exclusively of the weight of the piston, should not exceed 25 pounds,

thus
$$p_n = 0.1 \frac{P}{D\,L} < 25$$

and
$$D \times L > \frac{1}{250} P.$$

If $P = 450 \dfrac{\pi D^2}{4}$ it will be required that $\dfrac{L}{D} > 1.41$;

and if $P = 350 \dfrac{\pi D^2}{4}$ it will be required that $\dfrac{L}{D} > 1.09$. (114)

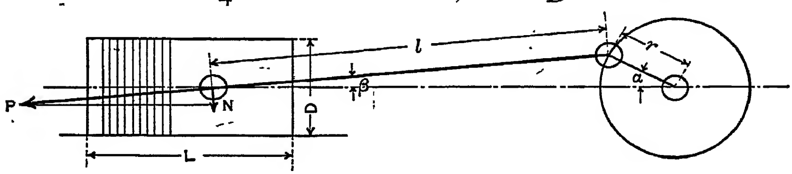


FIG. 79.

In a 20 × 30 engine with high compression we have

$$P = 314 \times 450 = 141{,}300 \text{ pounds},$$

and $N_{max.} = 0.068 \times 141{,}300 = 9{,}608$ pounds,

$D = 20$,

$L = 30$;

thus $F = 0.68\, D\, L = 408$ square inches.

Hence $p_n = \dfrac{9{,}608}{408} = 23.5$ pounds.

The Strength of the Piston.—In order to give stiffness to the piston it should be heavily ribbed, from the middle of the bottom some distance down its side, as in Fig. 80. The bending strain in the flat bottom, between the ribs and the circular wall, of a radius approximately ¼ D will be

$$S_b = \tfrac{1}{16}\, p \frac{D^2}{t^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (115)$$

when p is the pressure per square inch of the piston, D its diameter in inches, and t the thickness of its bottom.

The proper bending strain to allow in this case, when the pressure per square inch of the piston is figured $p = 450$ pounds, would be $S = 3{,}500$ pounds.

Thus, $$3{,}500 = \frac{1}{16}\,450\,\frac{D^2}{t^2},$$

or $$t = 0.09\,D.$$

Hence, the piston for a 20 × 30 cylinder requires a thickness $t = 0.09 \times 20 = 1\tfrac{7}{8}$ inch; the material being strained 3,500 pounds.

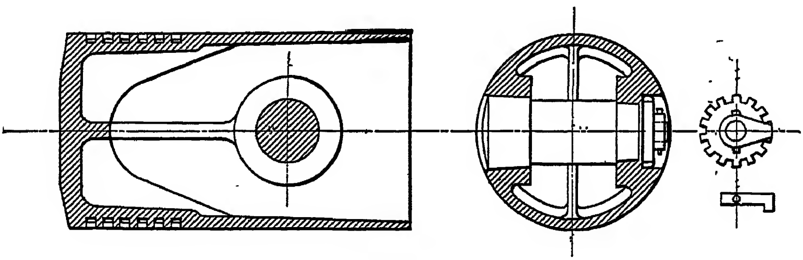


FIG. 80.

When the bottom is made curved to a radius $r = 2\,D$ the thickness of the bottom may be made $t = 0.075\,D$.

The Strength of the Piston Pin.—The pins for large pistons are most conservatively figured, for strength, as if supported at the mid-sections of their bearings in the trunnion bosses. Considering the load from the connecting-rod to be evenly distributed over the pin, we get, according to Fig. 81, the bending moment.

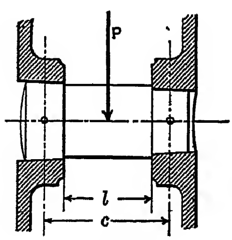


FIG. 81.

$$M_b = \frac{P}{2}\left(\frac{c}{2} - \frac{l}{4}\right).$$

The section modulus is $0.1\,d^3$, hence the maximum strain at the middle of the pin

$$S_b = 2.5\,\frac{P}{d^3}\left(c - \frac{l}{2}\right). \quad . \quad . \quad . \quad . \quad (116)$$

For a 20-inch engine:

$P = 141{,}300$ pounds,
$d = 7\tfrac{1}{2}''$,
$l = 9\tfrac{1}{2}''$,
$c = 13\tfrac{3}{8}''$.

Thus $$S_b = 2.5 \frac{141{,}300}{422} (13.375 - 4.75);$$

$$S_b = 7{,}200 \text{ pounds.}$$

The Piston Ring.—The open-end piston is generally provided with a number of spring-rings in order to reduce the pressure with which each ring must be sprung against the cylinder; 5 to 7 rings are often employed, exerting, each, a pressure against the cylinder of 4½ to 3½ pounds per square inch.

The thickness of the ring and the amount which has been cut out of the complete ring is of course what determines its stiffness when in place.

When a cast-iron ring, D inches in diameter, is sprung together a distance e the maximum strain in the middle section will be

$$S_b = \frac{e\ t\ E}{2.4\ D^2}; \qquad \ldots \ldots \qquad (117)$$

t being the thickness of the ring, and E the modulus of elasticity, for cast-iron = 12,000,000. And the tension per square inch of ring-surface will be

$$q = \frac{S_b}{3} \left(\frac{t^2}{D^2}\right),$$

or

$$S_b = 3\ q \frac{D^2}{t^2}.$$

If we allow a strain S_b = 11,000 pounds, and require the spring-tension to be q = 3½ pounds per square inch, we obtain

$$t = \frac{D}{32}.$$

In springing the ring over the piston there will be exerted a bending strain in the fibres

$$S_b = \frac{E\ t^2}{2.5 \times 0.25\ D^2} = 19{,}200{,}000 \frac{t^2}{D^2},$$

which for $t = \frac{D}{32}$ becomes

$$S_b = 19{,}000 \text{ pounds.}$$

The width of the ring is made, for small engines, ⅜ to ½ inch, and ⅝ to ¾ for large ones.

Fig. 80 shows the general construction of an open-end piston for a cylinder of medium size. The piston-pin nut is slotted for a spanner, and it is locked by means of a keeper, which is fitted to a spanner notch, and secured to the wrist-pin by means of a taper pin driven through the eye of the keeper and through a prolongation of the wrist-pin proper.

The piston should be fitted $\frac{1}{1000}$ inch free in the cylinder, per inch diameter, for its general length, and its extreme inside end is generally turned off $\frac{1}{64}$ to $\frac{1}{32}$ inch free, tapering to about 1½ to 3 inches forward. The smaller figures apply to the case of smaller pistons and the larger ones to medium sizes, of 12 to 20 inches in diameter.

The Connecting-Rod.—The main body of the connecting-rod is, for strength, treated as a hinged strut subjected to a compressive force equal to the total pressure on the piston. The load it can safely carry is computed according to the formula

$$fP = \frac{\pi^2 E J}{l^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (118)$$

in which

P is the safe compressive load;
f the factor of safety;
E the modulus of elasticity $= 30{,}000{,}000$ for steel;
J the moment of inertia of the middle section of the rod

$$= 0.05\ d^4 \text{ for a circular rod}$$

$$= \frac{b\ h^3}{12} \text{ for a rectangular section;}$$

d the diameter of the middle section, in inches;
l the length of the rod, in inches.

For a 20 × 32 engine

$$P = 141{,}300,$$

$$J = 0.05 \times 4.75^{-4} = 25.5,$$

$$l = 80.$$

Hence,
$$f = \frac{9.86 \times 30{,}000{,}000 \times 25.5}{141{,}300 \times 6{,}400};$$

$$f = 7.5.$$

Wrist-Pin End of Rod.—The force required for the acceleration

of the piston and piston-pin at the head-end centre causes the maximum tensile strain in the eye of the wrist-pin end of the rod.

If the weight to be accelerated is G_2, we have, according to equation 101,

$$P_2 = 0.000034\, G_2\, N^2\, r.$$

P_2 being the accelerating force at the head-end centre,

N the number of revolutions per minute, and

r the crank-radius in inches.

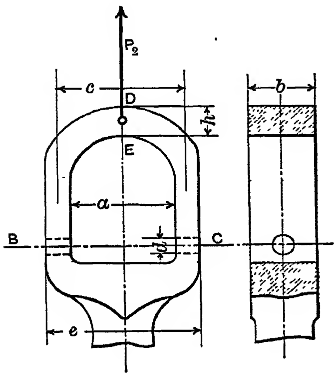


FIG. 82.

The area of the cross-section through the eye of the rod must be made to resist, in tension, the force P_2.

It will not, generally, be necessary to strain the material higher than at 5,000 pounds.

The bending strain on the back of the eye, at section $D\,E$, Fig. 82, will be:

Maximum bending moment $M_b = \dfrac{P_2\, c}{4}$,

Section modulus $\dfrac{J}{a} = \dfrac{b\, h^2}{6}$,

Maximum bending strain $S_b = \dfrac{1.5\, P_2\, c}{b\, h^2}$.

In a 20 × 32 engine we have:

Weight of piston	730 pounds.
Weight of piston-pin	180 pounds.
Total	910 pounds.

Hence $P_2 = 0.000034 \times 910 \times 25{,}600 \times 16$, at 160 revolutions.

$P_2 = 12{,}670$ pounds.

Other data are:

$a = 9\tfrac{1}{4}''$, $b = 6\tfrac{1}{4}''$, $c = 11''$, $d = 1\tfrac{3}{8}''$, $e = 13\tfrac{1}{4}''$, $h = 2\tfrac{3}{4}''$.

Thus the area of section $B\,C$

$$F = (e - a)\,(b - d)$$
$$= (13.25 - 9.25)\,(6.25 - 1.375) = 19.5 \text{ square inches.}$$

The tensile strain in this section, therefore,

$$S = \frac{P_2}{F} = \frac{12,670}{19.5} = 650 \text{ pounds.}$$

The bending strain at $D\,E$ is

$$S_b = \frac{1.5 \times 12,670 \times 11}{6.25 \times 7.56} = 4,420 \text{ pounds.}$$

Crank-End of Rod.—The maximum strain on the crank-end cap and cap-bolts is due to the force required for the acceleration of the reciprocating parts when the crank passes the head-end centre.

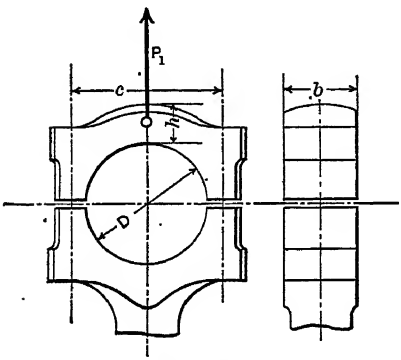


Fig. 83.

If G is the weight of the reciprocating parts (the piston and piston-pin and one-half of the connecting-rod), the accelerating force at the head end centre is

$$P_1 = 0.000034\, G\, N^2\, r;$$

N being the number of revolutions per minute and
r the crank radius in inches.

If the area at the bottom of the thread of two cap-bolts is $2a$, then the maximum tensile strain is $\frac{P_1}{2\,a}$ per square inch.

The bending strain in the rod-end cap:

If the dimensions of the cap are as shown in Fig. 83, the bending moment due to P_1, evenly distributed, becomes

$$M_b = \frac{P_1}{4}\left(c - \frac{D}{2}\right).$$

The section modulus for the middle section of the cap is

$$\frac{J}{a} = \frac{b\,h^2}{6}.$$

The maximum bending strain in cap

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{1.5\ P_1}{b\,h^2}\left(c - \frac{D}{2}\right). \quad . \quad . \quad . \quad (119)$$

For a 20 × 32 engine at 160 revolutions:

The weight of 20″ piston	730 pounds.
The weight of piston-pin	180 pounds.
½ the weight of connecting-rod	530 pounds.
Total	1440 pounds.

Hence $P_1 = 0.000034 \times 1{,}440 \times 25{,}600 \times 16 = 20{,}054$ pounds.

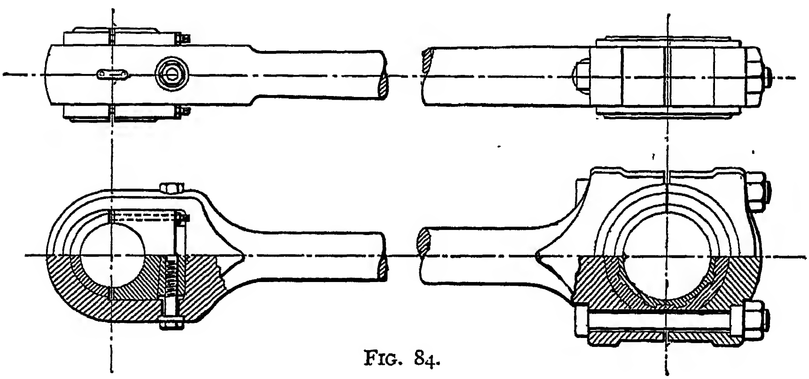


FIG. 84.

If two 2-inch cap-bolts be used (the area at bottom of thread 2 × 2.3 square inches), the maximum strain in the bolts due to inertia becomes

$$S = \frac{20{,}054}{4.6} = 4{,}360 \text{ pounds per square inch.}$$

Bending strain in the cap:

The distance between bolts,	$c = 12.75''$.
Bore of cap	$D = 12.5''$.
Thickness of cap	$h = 3.0''$.
Width of cap	$b = 6.0''$.

Maximum bending strain·

$$S_b = \frac{1.5 \times 20{,}054}{6 \times 9}(12.75 - 6.25) = 3{,}620 \text{ pounds.}$$

Should the shell bear on the cap at the middle only, the strain becomes $2 \times 3{,}620 = 7{,}240$ pounds.

A commonly employed design for a connecting-rod for medium-sized engines is shown in Fig. 84, and an alternate design of the wrist-pin end in Fig. 85.

The Strength of the Fly-Wheel.—Disregarding the influence of the arms on the strength of a fly-wheel ring, the tensile stresses

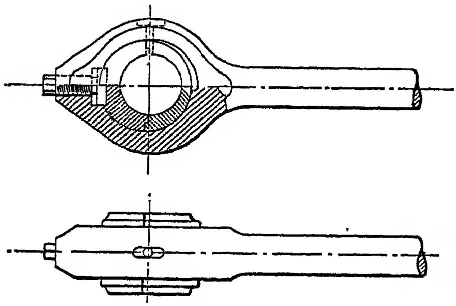


Fig. 85.

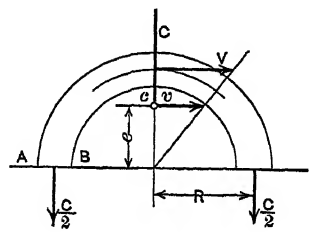


Fig. 86.

which act in each section, $A\ B$, of the ring, Fig. 86, are $\frac{1}{2}\ C$, when C is the centrifugal force due to one-half of the weight of the ring.

The centrifugal force, applied at the centre of gravity c, is

$$C = \frac{W}{2\,g}\ \frac{v^2}{e},$$

when W is the weight of the complete rim, in pounds, and v the velocity of the point c, in feet per second.

But as $\qquad e = \frac{2\,R}{\pi}, \quad$ and $\quad v = \frac{2\,\pi\,e\,N}{60},$

we get $\qquad C = \frac{W}{g}\ \frac{\pi\,R\,N^2}{900} = 0.00010848\ W\,R\,N^2,$

and the stress in each section of the rim.

$$\tfrac{1}{2}\ C = \frac{W}{2\,g}\ \frac{V^2}{\pi\,R} = 0.00005424\ W\,R\,N^2; \quad . \quad (120)$$

V being the velocity of the centre of the wheel-rim, in feet per second.

Assuming the rim-velocity to be 100 feet per second = 6,000

feet per minute, which is the highest velocity generally allowed in cast-iron wheels;

A the area of the rim-section in square feet ($= 144\,A$ square inches), then $W = 2\,A\,\pi\,R\,450$ pounds for a cast-iron rim, and

$$\tfrac{1}{2}\,C = \frac{4{,}500{,}000\,A}{g}.$$

Thus, the strain per square inch of the material is practically 1,000 pounds.

When a wheel is made in halves, the links joining the halves together should not be strained higher than 7,000 to 8,000 pounds per square inch, when they are submitted to direct tension only. The rule is, therefore, to make the smallest cross-section of the links for one joint 12½ to 14½ per cent of the section of the wheel-rim. When the links, or bolts, making the joints are not located centrally to the centre of gravity of the section, the bending moment acting in the joint must be considered, and the area of the bolts and links increased accordingly.

Fig. 87 shows the design for a wheel made in halves in which ordinary T-head links are used for shrinking the rim-halves together. In the hub are used bolts, and these should be made of a liberal diameter, as, in shrinking them in place, they are put to a strain which may be considerable. The diameters of the hub-bolts given in Table XXVIII have been found satisfactory in practice.

One key only should be used in the hub, as it is much easier to fit one large key well than to fit two smaller ones.

Tables of Practical Data Pertaining to Four-Cycle Single-Acting Gas-Engines.—The data contained in the following Tables, XXV and XXVI, referring to the power and dimensions of a line of medium-size producer-gas engines are revised from a line of engines in successful operation. The power and principal data given may be considered quite conservative.

Table XXV contains data with reference to the power, speed, and expected efficiency.

Table XXVI contains the principal dimensions of centre-crank shafts, the general design of which is shown in Fig. 88.

Tables XXVII and XXVIII give the weight and principal

TABLE XXV.

Dimensions and Power of Producer-Gas Engines.

Piston Diameter	12	13	14	15	16	17	18	19	20	21	22
Stroke	18	20	20	24	24	28	28	28	32	32	32
Revolutions per minute	220	210	210	200	200	180	180	180	160	160	160
Piston speed, feet per minute	660	700	700	800	800	840	840	840	853	853	853
Piston area, square inches	113.1	132.7	153.9	176.7	201.1	227.0	254.5	283.5	314.2	346.4	380.1
M.E.P	69	70	70	70	70	71	71	71	72	72	72
Maximum I.H.P.	39	49	57	75	85	102	115	128	146	161	177
Maximum B.H.P.	32	40	47	63	71	86	98	108	124	137	150
Rated B.H.P.	28	35	41	55	62	75	85	95	110	120	130
Mechanical efficiency	83	83	83	84	84	85	85	85	85	85	85

Maximum I. H. P. $= \frac{P_{mc} L A N}{2 \times 33{,}000}$. Maximum B. H. P. $= m \frac{P_{mc} L A N}{2 \times 33{,}000}$.

TABLE XXVI.

Dimensions of Centre-Crank Shafts.

Size of Engine	d	l	d_1	l_1	d_3	l_3	d_4	b	h	c
12×18	5½	5½	5	9½	5	10	6½	3½	7	12½
13×20	6¼	6¼	5½	11	5½	11	7	4	7½	13¾
14×20	6¼	6¼	5½	11	5½	11	7	4½	8	14¾
15×24	7	7	6½	13	6½	13	8	4½	8	15¾
16×24	7	7	6½	13	6½	13	8	4½	8	15¾
17×28	8½	8½	8	15	8	16	9½	5½	9	18¾
18×28	8½	8½	8	15	8	16	9½	5½	9	18¾
19×28	8½	8½	8	15	8	16	9½	5½	9	18¾
20×32	10	10	9½	18	9½	19	11	6½	10½	21½
21×32	10	10	9½	18	9½	19	11	6½	10½	21½
22×32	10	10	9½	18	9½	19	11	6½	10½	21½

The letters at head of columns refer to dimensions as denoted in Fig. 88.

TABLE XXVII.

Diameter, Speed, and Weight of Fly-Wheels.

Size of Engine.	Rated Brake Horse Power.	Number of Revolutions.	Diam.	Rim-Speed.	Weight of "Standard Wheel" $K = 35$.			Weight of "Extra Heavy Wheel" $K = 70$.		
					Rim. Lbs.	Hub and Arms. Lbs.	Total. Lbs.	Rim. Lbs.	Hub and Arms. Lbs.	Total. Lbs.
12×18	28	220	7′–6	5183	2400	1500	3900	4800	1600	6400
13×20	35	210	8′–0	5277	3100	1600	4700	6200	2000	8200
14×20	41	210	8′–0	5277	3600	1600	5200	7200	2000	9200
15×24	55	200	9′–0	5654	4700	2600	7300	9400	3000	12400
16×24	62	200	9′–0	5654	5300	2600	7900	10600	3000	13600
17×28	75	180	10′–0	5657	6400	3500	9900	12800	4400	17200
18×28	85	180	10′–0	5657	7200	3500	10700	14400	4400	18800
19×28	95	180	10′–0	5657	8100	3500	11600	16200	4400	20600
20×32	110	160	11′–0	5530	10100	5200	15300	20200	6400	26600
21×32	120	160	11′–0	5530	11100	5200	16300	22200	6400	28600
22×32	130	160	11′–0	5530	12200	5200	17400	24400	6400	30800

TABLE XXVIII.

Principal Dimensions for the Wheels of Table XXVII.

Diameter.	Standard Wheel.						Extra Heavy Wheel.						Hub.			
	Weight.	*h*	*w*	*a*	*b*	No. of Arms.	Weight.	*h*	*w*	*a*	*b*	No. of Arms.	*c*	*d*	*l*	Bolt Diam.
7′-6	3,900	6	6	7½	3¾	6	6,400	8½	8½	8	4	6	6½	14	13	1¾
8′-0	4,700	6¼	7	8	4	6	8,200	9	9¼	9½	4¾	6	7	15	14	1¾
8′-0	5,200	6¼	8	8	4	6	9,200	9	11½	9½	4¾	6	7	15	14	1¾
9′-0	7,300	7½	8	9	4½	6	12,400	10	12¼	10	5	6	8	18	16	2
9′-0	7,900	7½	8¾	9	4½	6	13,600	10	13	10	5	6	8	18	16	2
10′-0	9,900	8½	8½	10	5	6	17,200	11	13¼	12	6	6	9½	21	19	2½
10′-0	10,700	8½	9½	10	5	6	18,800	11	15	12	6	6	9½	21	19	2½
10′-0	11,600	8½	10½	10	5	6	20,600	11	16½	12	6	6	9½	21	19	2½
11′-0	15,300	9½	10½	11½	5¾	6	26,600	12	16½	13½	6¾	6	11	24	22	2¾
11′-0	16,300	9½	11½	11½	5¾	6	28,600	12	18¾	13½	6¾	6	11	24	22	2¾
11′-0	17,400	9½	12½	11½	5¾	6	30,800	12	20	13½	6¾	6	11	24	22	2¾

A suitable taper of the arms is:
For the wide side ⅜ inch per foot, over all.
For the narrow side $\frac{3}{16}$ inch per foot, over all.

The letters at head of columns refer to dimensions as denoted in Fig. 87.

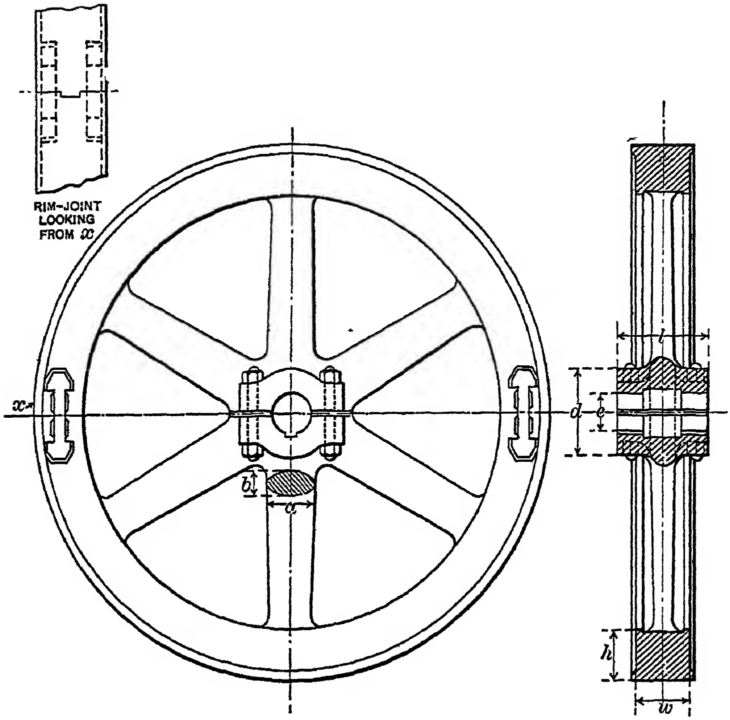


Fig. 87.—Fly-Wheel. For principal dimensions for various classes of engines see Tables XXVII and XXVIII.

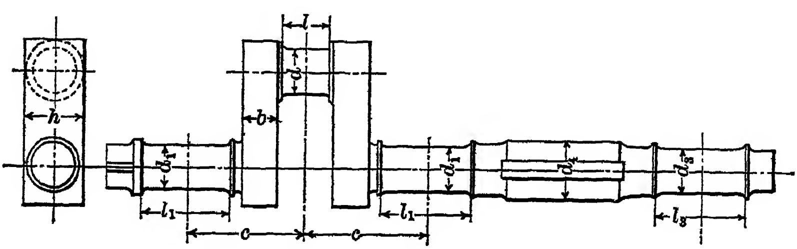


Fig. 88.—Centre-Crank Shaft. For dimensions suitable for various sizes of engines see Table XXVI.

dimensions of two types of fly-wheels generally used in gas-engine practice. They are designated by the names "Standard Wheels" and "Extra Heavy Wheels." The former are suitable for general mill service (the coefficient of steadiness $K = 35$); the latter is required for electric light and power service ($K = 70$).

The sizes of crank-pins and piston-pins, Table XXIX, have been determined by the methods explained in the preceding, and they conform closely with good general practice.

The bearing-pressures on the pins contained in the 6th and 10th columns of the table are the maximum pressures that occur generally only under very good conditions, corresponding to 450 pounds per square inch of the piston. The pressure on the piston under ordinary conditions is, on an average, not over 350 to 400 pounds, and, hence the bearing-pressures on the pins corresponding to these figures are $\frac{7}{9}$ to $\frac{8}{9}$ of those given in the table.

As the strength of all pins must be figured to suit the maximum pressure that is likely to occur, and the bearing-pressure on the pins being generally determined in connection with their strength, both the strength and bearing-pressure per square inch projected surface have been referred to the maximum pressure 450 pounds per square inch of the piston.

Automatic Valves.—Only small unimportant engines have inlet-valves automatically operated by the suction of the piston, as the simple means whereby the valves can be operated mechanically amply justifies its adoption. It is evident that automatically operated valves must be particularly unsuitable for engines running at a high speed, because the higher the speed, and the quicker the valve must open and close, the heavier springs it is necessary to apply back of the valves, to overcome the inertia in moving them. The pressure difference between the inside of the cylinder and the outside supply must, therefore, in high-speed engines, be considerable before the valves lift. This causes loss, both in efficiency and in capacity of the engine. The former since the piston must move against a considerable suction-resistance, the latter due to the low density of the charge in the cylinder at the completed suction-stroke. Engines built with the object in view of obtaining the highest possible power in a

TABLE XXIX.

Size of and Bearing Pressure on Crank-Pins and Piston-Pins.

Size of Engine.	Total Pressure on Piston at 450 lbs. per sq. in.	Crank-Pin.				Piston-Pin.			
		Diameter.	Length.	Projected Area.	Pressure per sq. in.	Diameter.	Length.	Projected Area.	Pressure per sq. in.
12 x 18	50,800	5½	5½	30.25	1,680	4½	5½	24.75	2,050
13 x 20	59,800	6¼	6¼	39.00	1,530	4¾	6½	30.87	1,930
14 x 20	69,300	6¼	6¼	39.00	1,770	4¾	6½	30.87	2,240
15 x 24	79,600	7	7	49.00	1,630	5½	7½	41.25	1,930
16 x 24	90,400	7	7	49.00	1,840	5½	7½	41.25	2,190
17 x 28	102,000	8½	8½	72.25	1,410	6¼	8½	53.12	1,920
18 x 28	114,000	8½	8½	72.25	1,580	6¼	8½	53.12	2,140
19 x 28	128,000	8½	8½	72.25	1,770	6¼	9	56.25	2,280
20 x 32	141,000	10	10	100.00	1,410	7½	9½	71.25	1,980
21 x 32	155,000	10	10	100.00	1,550	7½	10	75.00	2,060
22 x 32	171,000	10	10	100.00	1,710	7½	10½	78.75	2,170

cylinder of small diameter, even if the efficiency should be of minor importance, are, therefore, always provided with mechanically moved valves.

To ascertain that the valve-setting is correct and that the valve-ports are adequate for the speed at which an engine is operating, weak-spring cards, that show the vacuum created on the suction-stroke and the back pressure during the exhaust, are generally taken in connection with engine tests. Fig. 89 is such a card taken with a 20-pound spring. The compression line starts from the end of the suction line *a a* at an initial pressure $1\frac{3}{8}$ pounds below the atmospheric line *A B* and it discontinues at *b*, which is as far as the instrument allows the spring to be compressed. The expansion line is not indicated in the card, excepting the very lowest end of it; the release taking place soon after the line becomes visible at *c*. The exhaust-line *d d* is approximately 2 pounds above the atmospheric line.

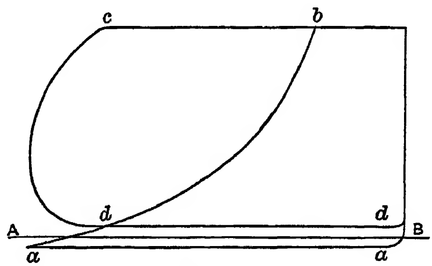


FIG. 89.

Inlet- and Exhaust-Valves.—The variation in the velocity

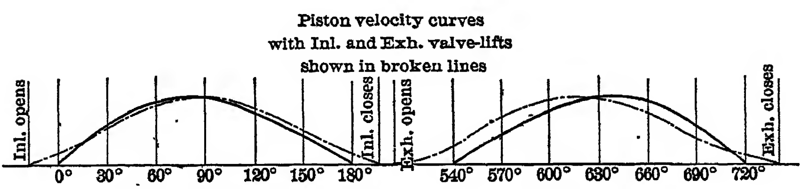


FIG. 90.

of the piston during the forward and return stroke is shown by the curve, Fig. 90. This curve has been obtained by plotting at the successive crank-positions the length of the ordinates given by the equation for the velocity of the piston-pin (see page 471).

$$S = V\left(\sin a + \tfrac{1}{2}\,\frac{r}{l}\,\sin 2\,a\right),$$

in which

S is the piston-velocity, in feet per second;

V the linear velocity of the crank-pin, in feet per second $= \frac{2\pi r n}{60}$;

a the crank-angle, counted from the head-end centre;

$\frac{r}{l}$ the ratio between the crank-radius and the length of the connecting-rod.

For any usual ratio $\frac{r}{l}$ the factor with which the crank-pin velocity must be multiplied to give the piston-velocity at the successive crank positions becomes:

a		$\left(\sin a + \frac{1}{2}\frac{r}{l}\sin 2a\right)$		
		$\frac{r}{l} = \frac{1}{5}$	$\frac{r}{l} = \frac{1}{5.5}$	$\frac{r}{l} = \frac{1}{6}$
0	360	0	0	0
30	330	0.587	0.579	0.572
60	300	0.954	0.946	0.939
90	270	1.000	1.000	1.000
120	240	0.78	0.788	0.794
150	210	0.413	0.421	0.428
180		0	0	0

In any case when, for practical purposes, it becomes necessary to construct the piston-velocity curve, and the exact ratio $\frac{r}{l}$ is not definitely known, the intermediate values, for $\frac{r}{l} = \frac{1}{5.5}$, can safely be used, as the discrepancy, if any, is very unimportant.

It would, of course, be desirable to have the inlet- and exhaust-valves lift and seat uniformly with the increase and decrease in the piston-velocity, in which case the velocity of the gas through the valve-ports would be constant. This they would do, practically, if given a motion as if actuated by a crank which passes the centres in unison with the main crank.

The required form of the valve-cams, to give the correct crank-motion, would then be that shown by the construction, Fig. 91.

The valves do, however, never actually begin to open when the engine-crank passes the centres, and for practical reasons the valve-cams are generally given a simpler form than that shown in Fig. 91, and on these accounts the increase and decrease in effective valve-area does not follow uniformly the increase and decrease in piston-velocity. The curve, according to which the actual opening and closing of the valves normally occur in practice, is constructed in Fig. 90, in broken lines, and it will be noticed that it follows only approximately the rise and fall of the piston-velocity curve.

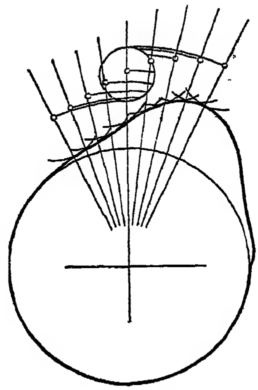


FIG. 91.

The curve representing the lifting of the valve is often, in practice, laid out in connection with the design of the valve-motion, in order to make sure that any improvement cannot be effected by a change in the form of the cams.

If the fluid velocity through the valve-ports be considered constant it is evident that the ratio between the maximum piston-velocity and the allowable maximum velocity of the gases in the valve-ports will determine the ratio between the full valve-area and the area of the cylinder. It is, however, more convenient, and more common, to determine the required valve-areas with reference to the mean piston-velocity, which is generally quoted, simply, as "the piston speed."

$$\text{The latter is} \quad P = \frac{2r}{\pi r} \times \text{maximum piston-velocity}$$

$$= \frac{1}{1 \cdot 57} \times \text{maximum piston-velocity.}$$

It is common practice to make the inlet- and exhaust-valves of sizes which call for fluid velocities through the valve-ports varying from 60 to 150 feet per second, counted with reference to the mean piston-velocity.

In small engines the valves can, without inconvenience, be made large, and they are to advantage often, in very small engines,

made as large as one-quarter of the cylinder-area. In large engines the valves and valve-springs become heavy, particularly when the valves are water-cooled, and for that reason the port-areas are cut down as much as can be done without incurring a considerable loss due to negative pressure on the piston. This it is particularly advisable to do with respect to large engines using cheap fuels as blast-furnace-gas or producer-gas, for the reason that a more reliable valve-mechanism will result. The inlet- and exhaust-valves are commonly made the same size. Sometimes, however, the exhaust-valve is reduced, so as to give an effective area only 80 per cent of the inlet-valve; this is done in order to reduce, as much as possible, the heavy strain in the valve mechanism required to open the exhaust-valve, against the pressure existing in the cylinder at the time for opening.

The following, Table XXX, gives the port-areas necessary, when it is desired to limit the fluid velocity in the ports to 60 feet, 100 feet, or 150 feet per second, as the case may be; the piston speeds being from 500 to 850 feet per minute. The last column of the table gives the required valve-diameter in terms of the diameter of the cylinder, with the assumption that the valve lifts enough so as to give the full area of the valve-port. This table is handy when proportioning port-areas to suit given conditions.

For instance, the table will show at a glance, that with a piston-speed of 500 feet per minute, and an assumed fluid velocity through the valve-ports of 100 feet per second, the required port-area becomes 8.33 per cent of the cylinder-area, or the valve-diameter 29 per cent of the cylinder-diameter. With a cylinder-diameter of, for instance, 6 inches, the valve would thus be only 1⅜ inches, which is very small. It will be better to limit the fluid velocity through the valve-ports to 60 feet per second, and make the valve-diameter 37 per cent of the cylinder diameter — 2¼ inches, or larger still if desired.

On the other hand, assuming the piston-speed to be 850 feet per minute, which is common in large engines, we obtain for a fluid velocity in the valve-ports of 100 feet per second a valve-area 14.2 per cent of the area of the cylinder, and a valve-diameter of 38 per cent of the cylinder-diameter. In the case of a 22-

inch cylinder, the valve would be 8⅜ inches, which is fully as large as it would be desirable to make it. As in this case the exhaust-valve, probably, would be water-cooled, a valve .34 per cent of the cylinder-diameter, calling for a fluid velocity of 140 feet per second, would, under most conditions, be preferable.

TABLE XXX.

Piston Speeds, Valve Port-Areas, and Gas-Velocities.

Piston Speed Ft. per min.	Vel. of Gases Through Valve-Port Ft. per sec.	Port Area in Per Cent of Cylinder Area.	Ratio Valve Diameter d to Cylinder Diameter D.
	60	13.9	$d=0.37D$
500	100	8.33	$d=0.29D$
	150	5.5	$d=0.23D$
	60	15.2	$d=0.39D$
550	100	9.2	$d=0.30D$
	150	6.1	$d=0.25D$
	60	16.6	$d=0.41D$
600	100	10.0	$d=0.32D$
	150	6.7	$d=0.26D$
	60	18.0	$d=0.42$
650	100	10.8	$d=0.33D$
	150	7.2	$d=0.27D$
	60	19.4	$d=0.44D$
700	100	11.5	$d=0.34D$
	150	7.7	$d=0.28D$
	60	20.8	$d=0.45D$
750	100	12.5	$d=0.35D$
	150	8.3	$d=0.29D$
	60	22.2	$d=0.47D$
800	100	13.3	$d=0.36D$
	150	8.8	$d=0.30D$
	60	23.6	$d=0.49D$
850	100	14.2	$d=0.38D$
	150	9.4	$d=0.31D$

Table XXX refers to mechanically operated valves, which are generally raised so as to give approximately the full opening of the valve-ports at its highest position.

The lift of a flat-seated valve corresponding to the full port-area is one-quarter of the valve-diameter, but, as inlet- and exhaust-valves are generally made with conical seats of approximately 45° angle, the lift should, theoretically, be somewhat more than that amount. However, the valve-seat being narrow, the area of the valve-stem may, in practice, be assumed to reduce the effective area to such an extent as to compensate for the discrepancy in the effective opening due to conical seat; and, in reality, the valves are seldom made to lift more than one-quarter of the valve-diameter.

The inertia of the valve, which the valve-cam has to overcome in opening it, and the spring at its closing, is, as will be shown, directly proportional with the weight and height of lift of the valve. But, the weight increasing in a much higher rate than in the direct proportion to the diameter, it would not serve the purpose of reducing inertia, to reduce the necessary lift of the valve by increasing its diameter.

The automatic spring-loaded valve, which can be lifted but a small amount, must be made of larger diameter than the mechanically operated one. The area of the former is ordinarily made 50 per cent larger than that of the latter, under otherwise similar service.

The extreme ratios between the valve-diameter, d, and the cylinder-diameter, D, will be found, in practice, to be:

For mechanically moved valves $d = 0.26\,D$ to $0.4\,D$,
and for automatic valves $d = 0.32\,D$ to $0.5\,D$.

The mechanically operated valve is generally made much heavier than what would be actually required for strength, in order to exclude any possibility of the valve warping or springing due to the heat to which it is exposed.

With respect to automatic valves, the necessary thickness of the valve-disc is suitably figured according to the formula for the

strength of round flat plates supported around the edge, which may be written:

$$t = 0.5\, d \sqrt{\frac{P_{max.}}{S}}, \quad . \quad . \quad . \quad . \quad . \quad (121)$$

t being the thickness required, in inches;
$P_{max.}$ the pressure per square inch of the valve-opening, and
S the strain per square inch of the material.

It must be remembered, in regard to automatic valves, that the durability of the valve, and valve-seat, will be increased by making the valve as light as the strength of the material will allow.

Assuming $P_{max.}$ to be 450 pounds, and allowing a strain $S = 12{,}000$ pounds per square inch, the thickness becomes

$$t = 0.09\, d.$$

This is, however, as light as a steel valve can safely be made.

The Valve-Seat.—The width of the face of the valve-seat should be made approximately $0.05\, d$.

Hence, for large engines the valve face becomes approximately $\frac{7}{16}$ inch, and for small engines the valve face becomes approximately $\frac{5}{32}$ inch.

The precaution is generally, taken with respect to more important engines, to make the exhaust valve-seat in a removable bushing which can readily be renewed in case of necessity.

The Valve-Stem.—The valve-stem serves to guide the valve properly, and to transmit the pressure that is required to cause it to open. The resistance against the opening of the inlet-valve is, principally, the tension of the valve-spring and a pressure due to the inertia of the valve. The latter force only is transmitted through the valve-stem. The resistance acting on the exhaust valve-stem is the inertia, in moving the value, and the pressure in the cylinder at the time it is opened. When the valve is being opened the inertia pressure is zero, and the pressure in the cylinder is, as a maximum, 60 pounds per square inch.

The valve-stem guide should be of ample length so as to guide and seat the valve properly, and, when so guided, the resistance on the valve-stem becomes, in effect, a simple compressive strain. This strain does not, ordinarily, need to be allowed higher than,

say, 3,000 pounds per square inch of the smallest section of the stem. Hence, if δ be the diameter of the smallest section,

we get
$$\delta^2 = \frac{60}{3,000} d^2,$$
or
$$\delta = 0.14\, d.$$

δ is commonly the diameter at the bottom of the thread, in the end of the stem, and the main body of the stem will, of course, be increased a suitable amount for machining and fitting.

The Valve-Springs.—The valve-spring must fill two requirements: first, it must exert enough pressure on the valve, when closed, to prevent its opening on the suction-stroke; secondly, it must have the required force to close the valve promptly during the short time for closing.

The first requirement, which has reference principally to the exhaust-valve, calls for greatly differing spring-tension for different cases, depending on the manner in which the engine is governed. In the case of a throttling or cut-off engine, it is evident that the volume of the clearance-space will affect the amount of vacuum that can be obtained in the cylinder on the suction-stroke, when the engine is running light. In an engine having a clearance-space of, for instance, thirteen per cent of the total cylinder volume there would be obtained, possibly, a vacuum of 11 pounds below the atmosphere, whereas in an engine having a clearance of 25 per cent the vacuum would not be over 6 pounds below the atmosphere.

The weight of the exhaust-valve, which helps the spring to hold the valve closed, should, strictly, be deducted from the required spring-tension, but as ordinarily this weight is slight, generally not over $\frac{8}{10}$ of one pound per square inch valve-area, it may be neglected.

Accordingly, the exhaust valve-spring for throttling- or cut-off engines should have a tension, when in place, of 6 to 11 pounds per square inch valve-area; the latter figure referring to producer-gas or blast-furnace-gas engines with high compression.

In engines the regulation of which is effected by the admission of a proportionally increased volume of air at light loads (constant-

volume regulations), the vacuum never becomes as great as in the throttling engine, and hence the valve-springs can in such engines be made lighter.

Some hit-or-miss engines require exhaust springs of a tension of only 4 to 6 pounds per square inch valve-area.

Spring-Tension Required for Prompt Closing of the Valve.— In order to close the valve promptly, it is required that the spring should have force enough to overcome the inertia due to the maximum change in velocity of the valve in closing.

Assume that a valve, weighing W pounds, is forced to its seat, a distance h inches, in T seconds, with uniformly increasing and decreasing velocity, such as the valve-cam would allow.

The maximum acceleration that must be given the valve is, approximately,

$$\text{acceleration} = 1.57 \frac{2}{T^2} \times \frac{h}{12} \text{ feet per second,}$$

and the force to give this acceleration will be

$$F = \frac{W}{g} \times \text{acceleration} = \frac{W}{32.2} \times \frac{1.57\,h}{6\,T^2} = \frac{W\,h}{123\,T^2}. \quad . \quad (122)$$

From Figs. 92 and 93, which are normal designs for an inlet and an exhaust valve-cam, it will be seen that the closing and opening throws extend, each, over a little less than ⅛ of the total circumference of the cam. In order that the cam-roller shall follow the closing cam-throw, it is necessary, therefore, that the valve closes in a little less than ¼ of one revolution of the engine. Assume, to be on the safe side, that the valve shall close in ⅙ of one revolution. Hence, the time for one revolution being $\frac{60}{N}$ seconds when N is the number of revolutions per minute, the time required for the closing of the valve will be

$$T = \tfrac{1}{6}\frac{60}{N} = \frac{10}{N},$$

and

$$T^2 = \frac{100}{N^2}.$$

An ordinary mechanically moved valve (not water-cooled), weighs, approximately, 0.8 pounds per square inch valve-area.

Thus $W = 0.8 \times \dfrac{\pi d^2}{4} = 0.8\,a$, approximately,

and the lift of the valve is, ordinarily,

$$h = \tfrac{1}{4}\,d;$$

d being the diameter and a the area of the valve-opening.

Inserting the preceding approximate values of T_2, W and h, in equation 122, it becomes

$$F = \frac{d\,N^2}{61{,}500}\,a. \quad . \quad . \quad . \quad . \quad . \quad (122a)$$

This equation gives:

for $d = 8$ and $N = 200$, $F = 5.2\,a$;
and for $d = 3$ and $N = 400$, $F = 7.8\,a$;

which shows the probable limits for the value of F to be between 5 and 8 pounds per square inch valve-area. These pressures being less than the pressure required to hold the exhaust valve of high-compression throttling or cut-off engines closed during the suction-strokes at light loads, it is evident that, in the case of engines of the latter type, the exhaust valve-spring should, generally, be given a higher tension than that required for prompt closing—with exception in the case of high-speed engines with water-cooled valves, which require that the necessary spring-tension be ascertained by inserting the correct values of W, h and T in equation 122.

In the case of automatically operated inlet valves, which generally are made very light, a spring-tension of only ¾ to 1 pound will be required; the exact amount depending on the speed of the engine.

The Valve Cams.—When cams are used for actuating the opening and closing of the valves they should be given such a form as to strike and leave the roller in an easy manner. If this is done a noiseless valve-operation will be obtained. Figs. 92 and 93 show the commonly employed construction of the inlet- and exhaust-cams, with throws made of case-hardened steel. The cam-rollers, as is shown in the figures, clear, in their closed positions, the cams by a small amount. This clearance should

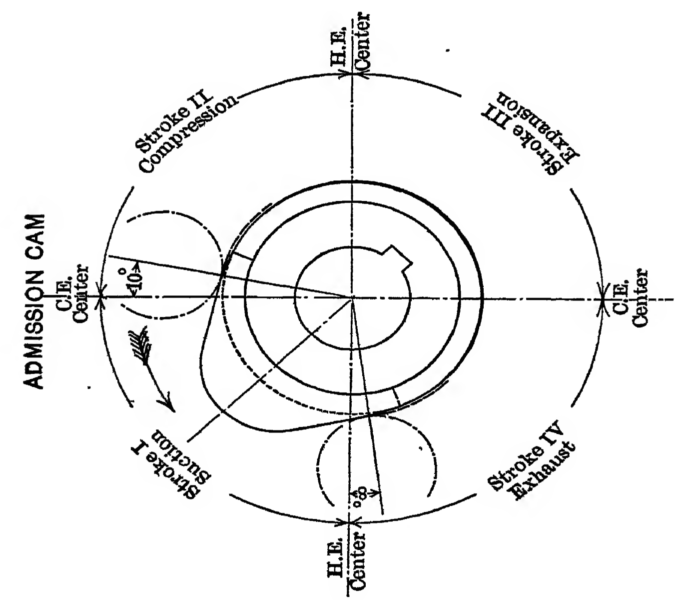


FIG. 92.

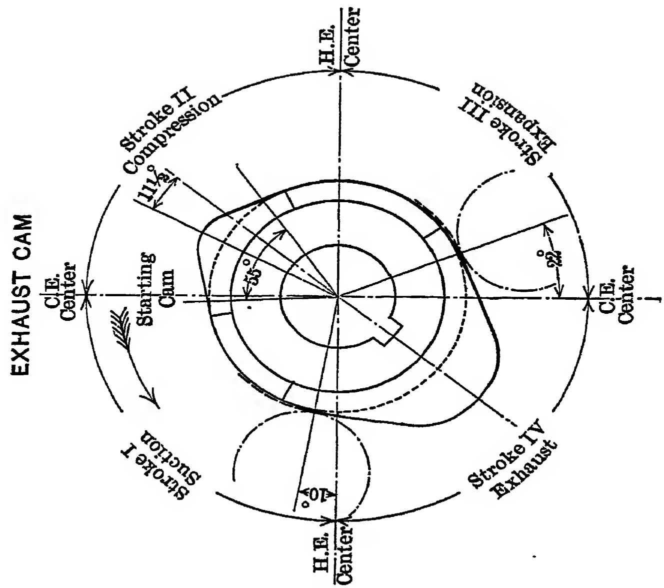


FIG. 93.

Diagrams of Valve-Setting for Producer-Gas Engines.

be carefully maintained after the valve-adjustment has been made, as a very small change will ordinarily throw the adjustment all out.

Valve-Setting.—The inlet-valve of a single-acting, four-cycle engine should properly open a little before the crank reaches the head-end centre, and close behind the crank-end centre. The exhaust-valve opens generally from 30 to 40 degrees before the crank reaches the crank-end centre, and closes 8 to 12 degrees behind the head-end centre. The exact setting of the valves must suit the speed of the engine and the nature of the fuel-gas used.

Figs. 92 and 93 are diagrams of the valve-setting used for producer-gas engines running at a speed of 800 feet per minute.

The Balancing of the Crank and Reciprocating Parts.—The simplest way to balance a revolving weight is by an equal weight placed on the opposite side of the shaft, and to balance, perfectly, reciprocating parts there must be used similar parts moving symmetrically in an opposite direction to those to be balanced. Vertically reciprocating parts can be perfectly balanced vertically, and horizontally reciprocating parts may be perfectly balanced horizontally by suitable revolving masses moving in an opposite direction to the reciprocating parts. In balancing reciprocating masses by revolving ones there will always, however, be evolved new unbalanced forces acting perpendicularly to the direction of motion of the reciprocating parts. Thus, in balancing vertically reciprocating parts there will result horizontal unbalanced forces, and in balancing horizontally reciprocating parts there will be evolved vertically unbalanced forces.

In balancing reciprocating parts it is of advantage to use a light weight placed as far as possible away from the centre of the axis, and when two wheels are used one-half of the total balance-weight may suitably be put at the periphery of each wheel. The whole balancing weight cannot be put in one wheel, excepting by locating another suitable weight on the opposite side of the centre-line of the engine. If this second balance-weight is not employed there will result a rocking force not normal to the centre-line of the engine.

In conformity with these principal facts regarding the balancing of engine-parts, it is customary, in an upright engine, to balance the revolving parts only, the crank-arms, the crank-pin and one-half the weight of the connecting-rod. A complete balancing of the reciprocating parts is not attempted, as it is better to have a free unbalanced force vertically, in which direction it can be resisted more effectively, than to change it into a horizontal unbalanced force.

In a horizontal one-wheel engine seven-tenths of the weight of both revolving and reciprocating parts are generally balanced.

Accordingly, if W_c is the total weight of the counter-weight;
r_c the radius at which it is applied, in inches;
r the radius of the crank, in inches;
W_1 the weight of the revolving masses, including the unbalanced weight of the crank-arms reduced to the radus r, the crank-pin and one-half the weight of the connecting-rod;
W_2 the weight of the reciprocating parts, including the weight of the piston and pin, and one-half the weight of the connecting-rod;

Then, for an upright engine

$$W_c = \frac{r}{r} W_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (123)$$

and for a horizontal engine

$$W_c = 0.7 \frac{r}{r_c} (W_1 + W_2). \quad . \quad . \quad . \quad . \quad (123a)$$

In large tandem engines it is often impossible to apply to the crank the full required balance-weight, and it becomes of advantage to apply part of the weight to the crank and part to the fly-wheel rim.

When two balance weights, on different distances from the centre of the engine, are employed, their relative weights are determined as follows:

In Fig. 94, W_1 is the weight to be balanced, acting at a radius r_1.

W_2 and W_3 are the balance-weights, located at distances l_2 and l_3 from the centre of the engine, and acting at radii r_2 and r_3.

The centrifugal force due to a weight W acting at a radius r inches, at N revolutions per minute, is

$$C = 0.0000285 \; N^2 r W.$$

In order to obtain balance between the forces C_1, C_2 and C_3 it will be required, first, that $C_1 = C_2 + C_3$;

thus, $$W_1 r_1 = W_2 r_2 + W_3 r_3, \quad . \quad . \quad . \quad . \quad (124)$$

and, secondly, that the moments relatively to O are zero;

thus, $$W_2 r_2 l_2 = W_3 r_3 l_3. \quad . \quad . \quad . \quad . \quad (125)$$

Hence, if the radii r_2 and r_3 and the distances l_2 and l_3 be

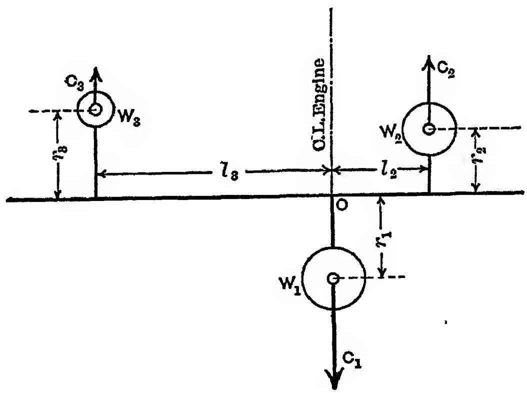


FIG. 94.

established, the weights W_2 and W_3 can readily be determined from equations 124 and 125.

Counter-Weight Bolts.—The bolts holding the counter-weight to the crank must be figured amply strong to resist the centrifugal force acting on it. This force is

$$C = \frac{12\, W_c V^2}{g\, r_c} = 0.0000285 \; N^2 r_c W_c. \quad . \quad . \quad (126)$$

W_c being the weight of the counter-weight,

r_c the distance from its centre of gravity to the axis of rotation in inches;

V the linear speed of the centre of gravity, in feet per second; and
N the number of revolutions.
g is the acceleration due to gravity $=32.16$.

Fig. 95 illustrates a common and reliable way to secure the

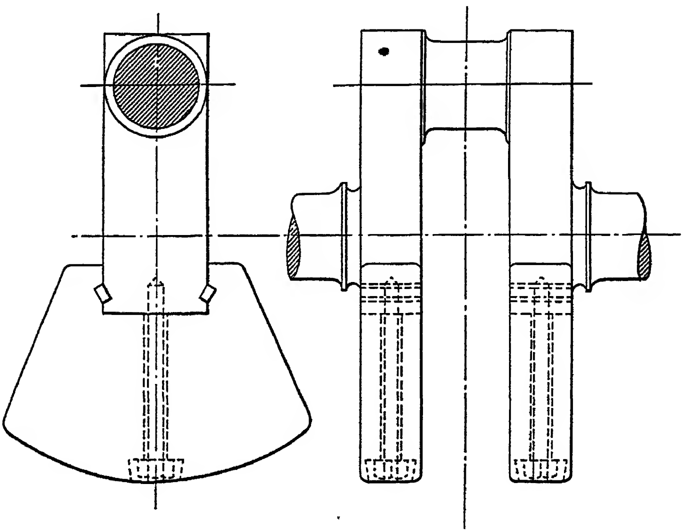


FIG. 95.

balance-weights to the crank-arms. The dovetailed nut pockets are, after the bolts are tightly in place, filled with lead to insure against the bolts unscrewing.

Water Cooling.—All gas-engines, excepting the very smallest,

FIG. 96.—Water-Cooled Piston.

which may be self-cooling, must be provided with some means for cooling the cylinder. Cylinders of a size up to 4 inches may be cooled by an air-current which is made to impinge on the

cylinder-jacket, preferably, then, provided with a ribbed cooling-surface to facilitate the dissipation of heat.

Cylinders over 4 inches are always water-cooled. The larger the cylinder, the more important it is to thoroughly cool the combustion-chamber, particularly around the exhaust valve-seat and spark-plugs. Faulty arrangements for the cooling of these parts, which are most liable to become over-heated on heavy loads, will cause pre-ignitions for compressions that otherwise would be safe and normal for the particular fuel used.

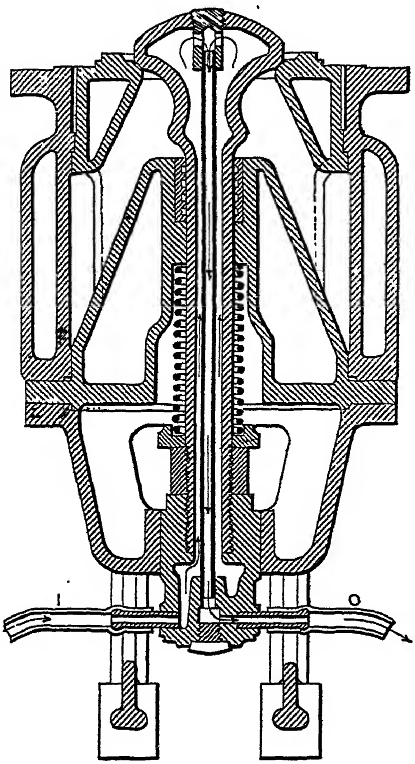


FIG. 97.—Water-cooled Exhaust-Valve.

Single-acting engines above 20 to 24 inches, and double-acting cylinders generally, are provided with water-cooled pistons and exhaust-valves; and the more liable the fuel is to cause pre-ignitions the more necessary it will be to cool these parts.

Fig. 96 shows a common method for cooling the piston and rod of double-acting engines. The cooling water is, by means of a telescoping tube, admitted to the hollow piston-rod, at the main cross-head, and it passes from there to the bottom part of the piston. As it becomes heated the water rises to the top of the piston and is discharged from the very highest point of it so as to, as thoroughly as possible, expel with it any air or steam that may tend to separate off. The hot water is drained from the rear cross-head, in a visible stream, by which its temperature and proper continuance can always be observed.

Fig. 97 illustrates a water-cooled exhaust-valve of modern type. The cooling water is admitted through a flexible hose, at *I*, and passes through the hollow valve-stem to the crown of the valve, from where it is discharged through a small central tube which stands in communication with the discharge hose *O*. From the discharge hose the water is drained, openly, into a drain funnel so that the flow can always be observed.

Cooling Water Required.—It may be stated, roughly, that as a maximum one and one-half times to twice as much heat is carried off by the cooling water as that converted into work. Accordingly, each horse-power corresponding to 2,545 heat-units per hour, the cooling water absorbs

$$(1\tfrac{1}{2} \text{ to } 2) \times 2{,}545 \text{ B.T.U. per hour per horse-power,}$$

or from 3,817 to 5,090 B.T.U. per hour per horse-power.

Each pound of the cooling water absorbs

$$(t_f - t_i) \text{ B.T.U.,}$$

when t_f and t_i are the temperature-limits between which it is heated; and, as the temperature of the jacket-discharge should ordinarily not be allowed to become over 185° F., it may be said, that approximately from 100 to 150 B.T.U. are absorbed per hour per pound of cooling-water.

Hence, the quantity of cooling-water ordinarily required per horse-power, will be between

$$\frac{3{,}817}{150} \quad \text{and} \quad \frac{5{,}090}{100};$$

from 25 to 51 pounds per hour,
or from 3.1 to 6.3 gallons per hour.

Allowing for any ordinary overload-capacity of the engine, a circulating pump supplying 8 gallons per each rated horse-power would be ample, provided the water is not supplied hotter than 85° F. For each 10° of temperature higher than 85°, at which the water is supplied, 10 per cent more pump-capacity will be required. It will be noticed that, if these requirements are filled, the pump-capacity will, ordinarily, be approximately 50 per cent above actual requirements.

The Double-Acting Cylinder.—A double-acting cylinder, as frequently carried out for the four-cycle engine-type, is shown in Figs. 98 and 99. The strains in the material of a cylinder of this construction, and which not infrequently have caused it to fail, are due to two separate causes: Strains due to the working pressure, and strains due to the expansion of the inside cylinder-wall, when heated by the working gases.

It is readily ascertained, that the part of the cylinder-barrel between the openings cut through it for the accommodation of the valves will without difficulty be made amply strong for sus-

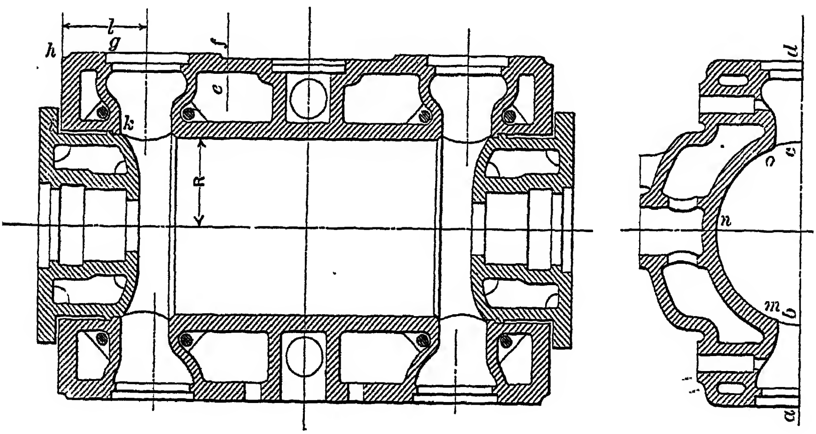


FIG. 98. FIG. 99.

Nuernberg Double-acting Cylinder.

taining the tangential strains due to the explosion pressure; figured as a plain cylindrical shell carrying internal pressure. Whether the axial strains, in a direction lengthwise of the cylinder, will be taken up by the inside barrel, or by the outside jacket, depends on the arrangement made to allow for the expansion of the inside wall.

The design of the cylinder illustrated is that used by the firm The Maschinen Gesellschaft Nuernberg, Germany, and its weak sections have, in practice, been found to be in the sections *a b* and *c d*, and also in the outside jacket, at section *e f*. Due to the openings for the valve-bonnets, which cut through the outside

and inside walls, it becomes necessary that the section $g\ h\ i\ k$ is of ample strength to sustain a total load $P_{max.}\ l\ R$, by which it will be strained, and on this account the distance $h\ i$ must be made of a considerable depth, or steel stay-bolts, as shown in the figure, may be used to help carry the load. Further, as the abutments for the arched part $m\ n\ o$ are cut away by the openings for the valve-bonnets, there will be caused transverse strains in the sections $a\ b$ and $c\ d$, that must be taken in consideration.

When the cylinder is made, as in the illustration, with the outside jacket-wall cast continuous from end to end, the question will arise, what effect the strains due to the expansion of the material will have on the strength of the construction as a whole.

It is reasonable to assume the temperature of the inside surface of the cylinder-barrel to be, on an average, 500° F., the cooling water 100°, and, hence, the average temperature-difference between the inside barrel and the outside jacket, approximately, 200° F. Such a temperature-difference would call for an expansion of the inside barrel of practically one-sixteenth of one inch, or more, relatively to the outside jacket-wall. The elasticity of the material will allow of this, assuming the water-space is made of proper depth. The Nuernberg construction is given adequate depth of water-space, but there will result in certain sections of the jacket a not inconsiderable tensile strain; and the inside cylinder-barrel will be, axially, in compression. The axial strains, therefore, due to the pressure on the piston will all be taken by the cylinder jacket, which, consequently, must be made of proper thickness and so ribbed that the axial strains become evenly distributed.

Some builders prefer to have the jacket core cut through the outside cylinder-wall at the middle of the cylinder, to avoid any longitudinal strain due to the expansion of the metal by the heat transmitted from the working gases. The cylinder-barrel will, then, take all the axial strain, which it fully can do, without increased stress, because the strain axially is only one-half as severe as that tangentially. The latter construction involves some little difficulty in making a water-tight expansion-joint all

around the cylinder, and this is what has been sought to be avoided in the Nuernberg construction.

The Piston-Rod Packing.—The main idea on which the construction of all modern rod packings is based is to reduce gradually, and as effectively as possible, from cell to cell of the packing, the pressure existing in the cylinder to that of the atmosphere; taking in consideration that ample provision must be afforded the rod to centre itself freely between its points of support at the

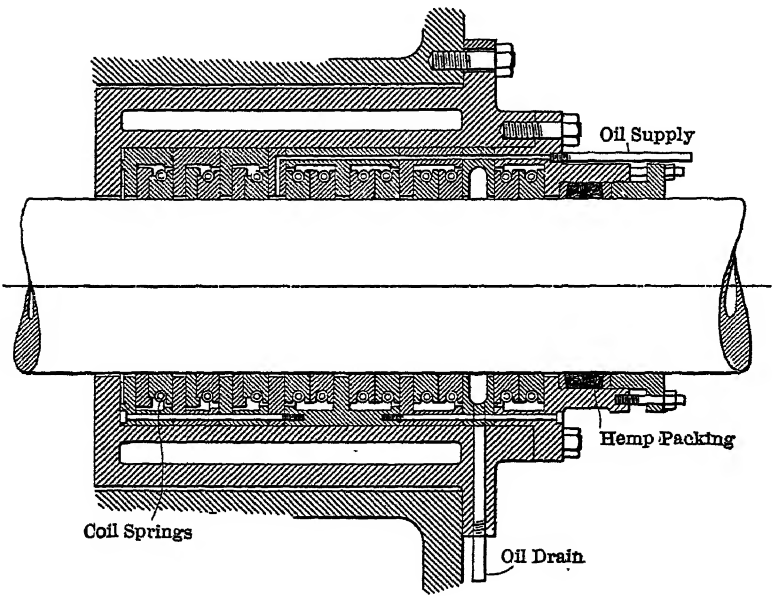


FIG. 100.—Schwabe Piston-Rod Packing.

main and at the rear crosshead. Of the construction of the rod-packings generally employed the Schwabe packing, Fig. 100, may serve as an illustration.

It consists of several pairs of three-parted soft cast-iron rings, breaking joints; the ring-sections being held together and sprung against the rod by suitable coil-springs. The packing is self-contained in a casing that can readily be taken apart, for the removal or examination of the rings. Effective lubrication and

cooling of the packing must be provided, and for the latter purpose the packing is, most generally, enclosed in a water-cooled packing-box, as seen in the figure. The oil-supply is admitted at a point from where the oil will most effectively be distributed without tending to leak too freely in to the cylinder. At the outside end of the packing there is provided a soft hemp-packing that will prevent the lubricant from escaping along the rod.

CHAPTER XII

GOVERNING

THE methods generally employed for controlling the speed of a gas-engine are: either the so-called hit-or-miss method or the throttling or cut-off method, or that of advancing or retarding the ignition.

Hit-or-Miss Governing.—In the first system the charge is cut out entirely for one or more pressure-strokes, whenever the speed becomes above the normal.

This is accomplished by the governor, by withdrawing a member, a pick-blade or a cam-roller, in the valve-actuating mechanism, causing the inlet-valve to remain closed against the admission of new charge. In case the inlet-valve is operated automatically by the suction of the piston, the governing may be actuated upon the exhaust-valve by holding it open during the admission-stroke, thereby preventing the spring-loaded inlet-valve from opening.

A great variety of hit-or-miss governing devices are in use, mostly in connection with small-sized engines where any particularly close regulation is not very essential. The governor proper may be a fly-ball governor or one of the inertia or pendulum type.

Various Forms of Hit-or-Miss Regulations.—Fig. 101 illustrates a form of hit-or-miss regulation controlled by a fly-ball governor, in combination with a pick-blade.

S is the cam-shaft, which generally is driven from the main engine-shaft by means of a pair of spiral gears, in such a manner that it makes one revolution while the main shaft makes two. The cam *C* secured to the cam-shaft, may thus be timed so as to actuate the cam-roller *D*, which is carried by the valve rocker *R*, at the proper time for the opening of the inlet valve *I*. The position of the pick-blade *P* is controlled by the governor, by means of the bell-crank *B*, so that, when the governor is running

below normal speed, the pick-blade is carried in a position to engage with the valve-stem, and open the valve; but when the speed is above the normal, the pick-blade will be in such a position as to miss the valve-stem, and the valve will remain closed.

Fig. 102 illustrates an inertia governor which regulates on the hit-or-miss principle by means of a pick-blade. It is a type used by the firm Crossley Bros., Ltd., Manchester, England, on small four-cycle engines. *S* is the cam-shaft and *C* the cam which

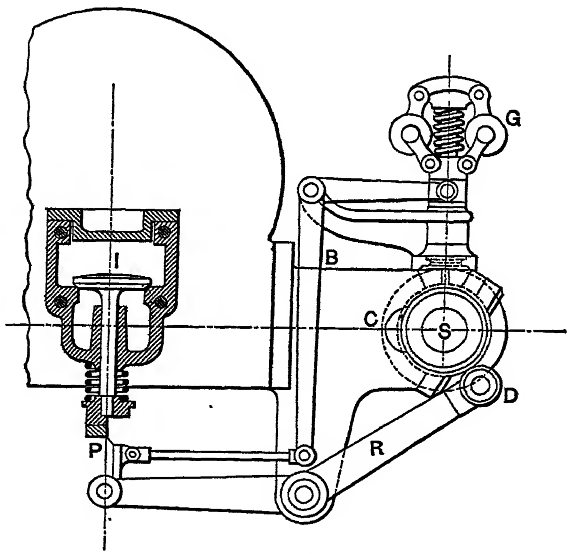


FIG. 101.

actuates the cam-roller *D* at the proper time for opening the inlet-valve. *A* is the governor arm, which on one end carries the weight *G*, and on the other the pick-blade *P*. The valve-rocker *R*, which transmits the motion from the roller *D* to the valve stem *I*, is fulcrumed at *F*, and carries the governor arm hinged on it at *H*. The governor arm, with the weight and pick-blade, is thus free to swing through a small arc relatively to the rocker *R*, excepting that the spring *M* holds it with a suitable force against the rocker, so as to permit the pick-blade, under certain circumstances, to meet the end of the valve-stem *I*, when the rocker is actuated

upon by the cam *C*. *K* is the valve-spring which holds the valve closed, and the spring *L* has for object to push the rocker *R* back to its neutral position.

The general motion of the governor-weight is along the arc *a* – *b* whenever the tension of the spring *M* is strong enough to hold it in this path. At a certain speed of the engine, however, the cam *C* will hit the roller *D* too swiftly for the spring to hold the weight in its regular path, and the inertia of the weight will then throw it back and carry the end of the pick-blade downward

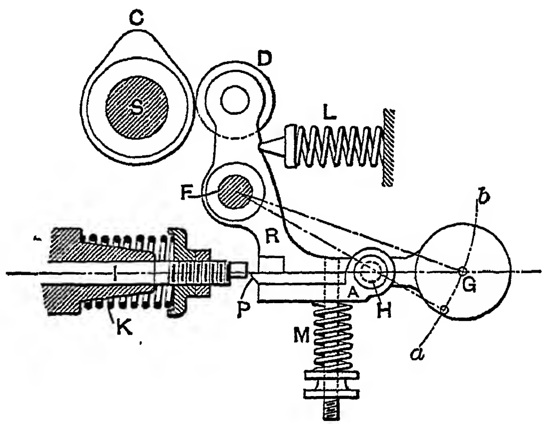


Fig. 102.

so as to miss the end of the valve-stem. When this happens the valve will remain closed against the admission of any charge.

Fig. 103 illustrates a so-called pendulum governor which occasionally is used for the regulation of small engines. *S* is the cam-shaft and *C* the cam which actuates the pushrod *P*. The latter is guided at *D* and *E*, and is carried back to its neutral position by the spring *L*. On a fulcrum-bracket, *B*, secured to the pushrod there is hinged a pendulum, *G*, which is made in one piece with the pick-blade arm *A*. *I* is the inlet valve-stem and *K* the valve-spring.

When the pushrod is given a reciprocating motion, in time with the required periods for opening and closing the valve, the pendulum will be set in a swinging motion, which at excessive

speeds will have for effect to carry the pick-blade out of contact with the end of the valve-stem and thus cause one or more miss-strokes. The spring *M* has for object to steady the pendulum and to reduce the arc of its oscillations, which, without it, would be excessive.

In Fig. 104 there are shown various arrangements of the pendulum-governor. In the arrangements *I* and *II* there are no retarding springs used, and as a consequence the swing of the pendulum and the sensitiveness of the regulation will be excessive. The arrangement *III* is practically the same as that of Fig. 103, and in the arrangement *IV* the valve will be hooked in and follow the motion of the push-rod, until, at excessive speeds, the mechanism becomes unhooked, leaving the valve to remain closed for one or more strokes.

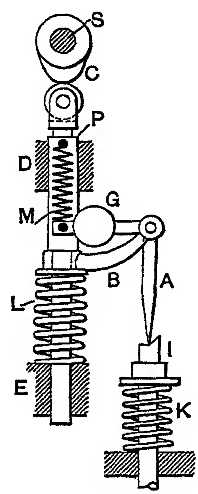


FIG. 103.

The Throttling or Cut-off Regulation.—In the throttling or cut-off regulating system the explosions and pressure-strokes follow each other always at regular intervals, but with diminished intensity at light loads, due to the admission of a charge of impoverished fuel-value. This system of governing, which is often applied to modern well-built engines, may be carried out according to two different methods. By the first, the so-called *constant-quality* method, the charge remains for all loads of a constant proportion of air and fuel, but at light loads it will be throttled, or cut off, during the suction-stroke to suit the load. The density of the charge at the end of the suction-stroke, as well as, consequently, its compression becomes, therefore, variable, and proportionate with the load.

By the second, the so-called *constant-quantity* method, the quantity of the charge remains uniform for all loads, but its quality is made poorer at light loads by means of the throttling or cutting off of the fuel alone; the air charge, strictly, must be increased by the amount the fuel is decreased, or, when the fuel is rich, it remains practically constant. As the density of the

charge after completed suction-stroke remains constant for all loads, therefore, the compression of the charge will also be constant under all load conditions.

When the fuel-charge alone is regulated by a cut-off valve, the object should be to admit the fuel as late during the suction-stroke as practicable, by which the advantage is sought, to obtain, as far as possible, a charge consisting principally of air near the piston, and which gradually grows richer in fuel nearer the igniter. For this purpose, the fuel-valve is opened late and closed near the end of the suction-stroke.

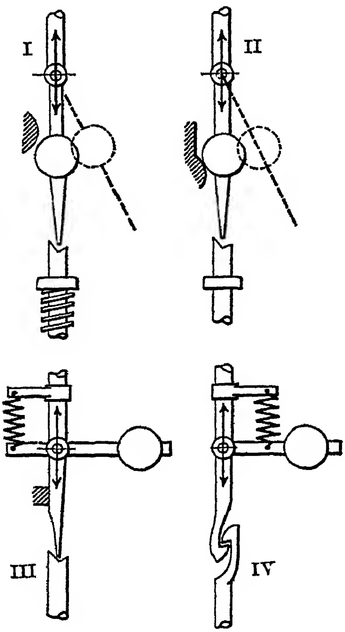


FIG. 104.

Constant-Quality Regulations.—The regulation system of the engine illustrated in cross-section in Fig. 105, consists of a butterfly valve *V*, controlled by the governor, by means of the governor-shaft *S*, the lever *L*, and the connection *C*. The gas arrives through the nozzle *N*, and the air through the opening *A*, into the mixing-chamber *M*. When the governor rises it closes the butterfly-valve more or less; thus, by throttling, it reduces the density of the charge so as to admit to the cylinder heating-valve in proportion to the momentary load requirements. This is a constant-quality throttling governing.

Instead of the governor throttling a charge of a constant mixture of gas and air by means of a butterfly-valve, a cut-off valve is sometimes used, which, under the control of the governor, cuts off the charge at a point of the stroke suitable to the load, similarly as in the steam-engine. An example of this class of governing is that of the Jacobson engine illustrated in Fig. 147, page 373.

Another type of constant-quality regulation is that employed by the Gasmotoren Fabrik Deutz, Cologne, Germany, and which is illustrated in Fig. 106. In this regulation the gas and air arrive to the admission valve through separate channels in which the gases are throttled to the correct proportions, and they are, at reduced loads, wire-drawn in the gas and air valve-ports, to the

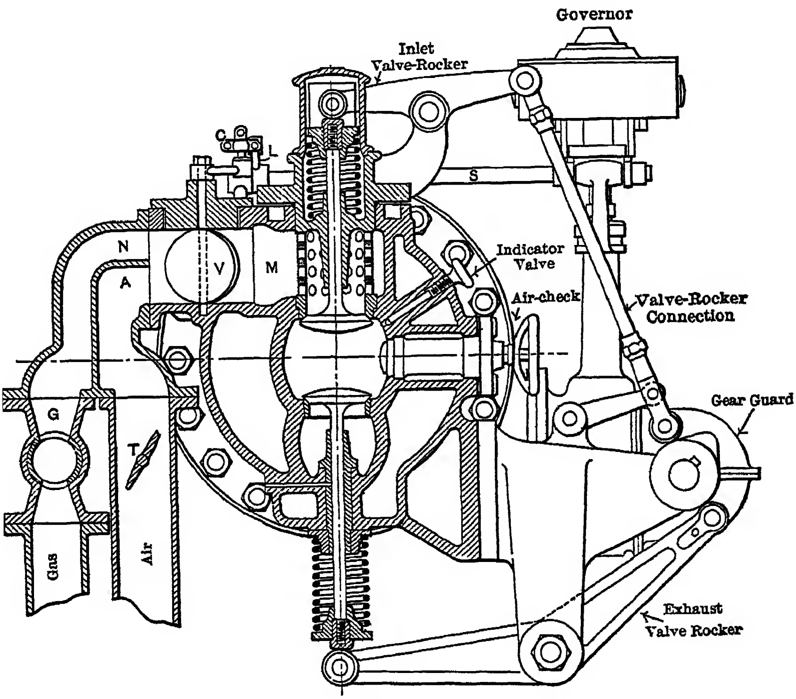


FIG. 105.

extent necessary for reducing the density of the mixture to correspond to the momentary load requirements.

Referring to the figure it will be seen that the air- and gas-valves, which are made in one piece, are slipped over the main inlet valve-spindle and held there between suitable collars. *R* is the admission cam-roller, which is actuated upon by the cam *C*, and which transmits motion to the valve-lever *L* by means of the valve-rod *D*. The valve-lever is not hinged to any fixed fulcrum, but a

movable roller, *M*, serves as fulcrum for depressing the admission valves. The position of the fulcrum-arm *A* is, as seen, controlled by the governor. For the position of the arm as shown in the figure the valves will obtain their maximum lift, but for a rising governor the fulcrum-roller *M* will move toward its inward position, at *E*, and thus, successively, increase the leverage with which the valves are actuated upon and cause their lift to be decreased. As the port-areas for the gas and air will always

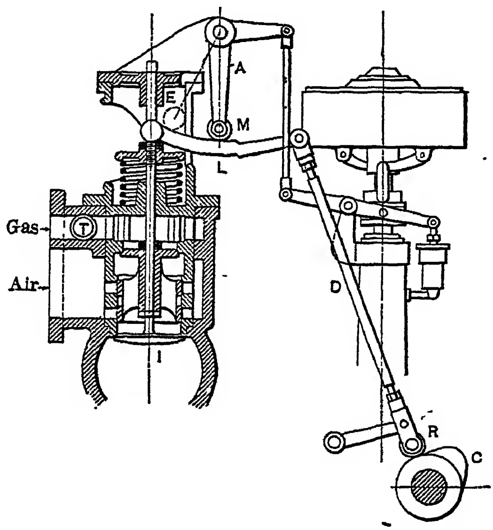


FIG. 106.—Deutz Admission Valve-Gear.

increase and decrease proportionately, therefore, the proportion in which the gas and air is admitted will remain constant.

T is a throttle in the gas-pipe by which the permanent proportioning of the gas and air is controlled.

If the charge actually remains of a constant quality by the use of the throttling or by the use of the cut-off regulations, there is then very little difference between the results derived by the two systems, because the charge must in either case be partially, or completely, excluded at one time or other during the admission-stroke, to the extent that the same quantity of charge is admitted; and the compression becomes in either case the same. Practically

there is, however, some difference between the two methods, because by the employment of a cut-off valve the quantity as well as the quality of the charge may be varied.

Constant-Quantity Regulations.—To obtain a constant-quantity regulation the gas must be throttled separately to suit the variations in the load. Such a regulation is illustrated in Fig. 107. The inlet-valve opens in this regulation under all load-conditions to its full lift, admitting always the full volume of charge, and while the gas may be throttled more or less the air

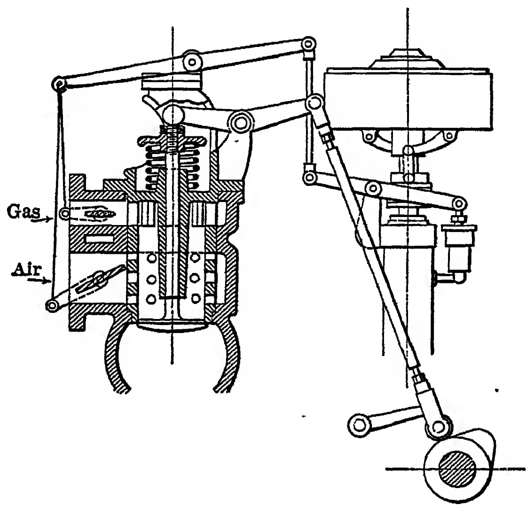


FIG. 107.—Constant-Quantity Regulation

charge is in a corresponding proportion increased. That is, when the gas-throttle is closed the air-throttle is correspondingly opened; both valves being under the control of the governor.

Fig. 161, page 397, represents a constant-quantity regulation in which the fuel-charge is regulated by means of the gas-valve, which is during the admission-stroke opened to a greater or less extent; the valve closing regularly at the end of the suction-stroke.

It is well known, and it will be shown presently by the results obtained from indicator cards taken at full and light loads, that the economy of the throttling or cut-off engine decreases rapidly

as the load decreases. The economy of the hit-or-miss engine, on the other hand, is more constant for all loads; as would be expected when all pressure-strokes are made under much the same conditions as to compression and mixture. The only dissimilarity between one power-stroke and another is in this engine due to the fact that after one or more misstrokes the cylinder becomes more thoroughly scavenged from the burned gases, and, perhaps, cooled by the air which is taken in and expelled during the misstroke.

In order, therefore, to attain as high economy as possible, under all load-conditions, the Crossley Bros., Ltd., Manchester, Eng., have, for years, been using a combination throttling and hit-or-miss regulation for engines running on variable loads. This regulation has also been made a feature of the Crossley engine manufactured in this country.

A new combination-regulation of this type was recently described by Mr. Atkinson, in a paper read before the Institution of Mechanical Engineers of London.

The regulation is shown in Figs. 108, 108*a* and 108*b*, and its principal feature is that, for loads above one-half the rated load of the engine, it acts on the throttling governing principle, whereas for loads below one-half the rated load it acts on the hit-or-miss principle. The latter feature is readily observed by a glance at the illustration. When, namely, the governor arm *L*, due to a light load, rises above a certain limit, it pulls the block *B* away from the path of the pick-blade, *P*, which is secured to the valve-rocker, *R*, and, as a consequence, the pick-blade passes below the block, thus failing to open the valve.

The inlet valve-stem is made hollow, and it is connected with a dash-pot plunger, *D*, moving air-tight in the vacuum-chamber, *C*. *S* is the valve-spring which holds the inlet-valve closed, and, the plunger and the valve being in one piece, the same spring serves also to push the plunger to the bottom of the vacuum-chamber. To allow the plunger to move to the bottom of the vacuum-chamber and close the valve promptly, there is, at *V*, a small snifting-valve opening outwards.

The force that opens the main valve when the pick-blade

engages with the block *B* comes through a stiff spring, *T*, from the push-head, *H*, which, normally, butts against a shoulder at the outside end of the hollow valve-stem; the spring *T*, thus, merely serving as a flexible valve-stem for opening the valve.

At *A* is a small air valve, which throttles the air-port leading to the vacuum-chamber, *C*. This valve is, by means of the connection *K*, controlled by the governor. When the governor-

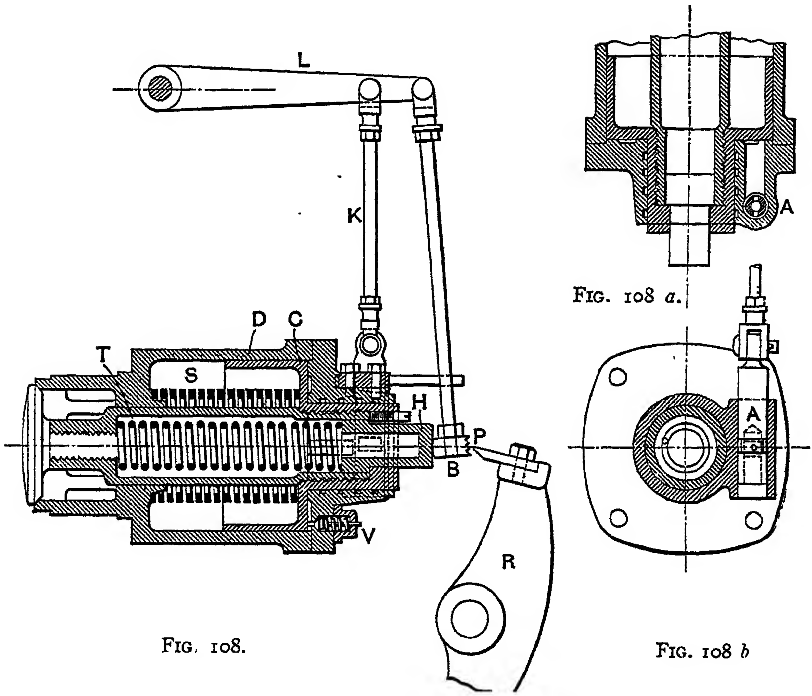


Fig. 108. Fig. 108 *a*. Fig. 108 *b*

arm swings low, at heavy loads, the valve *A* will open the air-port fully, thus preventing any vacuum from forming back of the plunger, *D*, and, hence, the inlet-valve will move freely to its full lift and admit maximum charge. When, however, the governor-arm swings high, at light loads, then the valve *A* will throttle the air-port leading to the vacuum-chamber, more or less, throwing a proportionally greater strain on the spring, *T*, in forcing the main valve open. This will, of course, have for effect to compress

the spring T, more or less, and to reduce the lift of the valve correspondingly.

Regulation by Retarding the Ignition.—In very small engines, temporary changes in the load may be taken care of without disturbing the regular working of the valves or altering the quality of the charge, simply by retarding the spark when a reduced effect is required. This system is of course wasteful, as the same quantity of fuel is consumed at light as at heavy loads, and it is used only when the actual efficiency of the engine is of minor importance.

The Governor.—The Hartung type of governor is, at present, very commonly used for the regulation of gas-engines of medium and large size. This governor, illustrated in Figs. 109 and 110, is of fly-ball type, using springs for counter-acting the centrifugal force of the weights, and it has very generally been adopted on account of its sensitiveness, minimum amount of internal friction and great power; it being one of the types that for the least weight of the fly-weights possess the greatest power. Another feature of this governor, that recommends itself strongly, is that the fly-weights can conveniently be enclosed in a casing which makes attention to the governor, when in motion, entirely safe, even in a crowded space. The governor is generally driven from the cam-shaft of the engine by means of a pair of spiral gears, or, occasionally, by means of spur-and-bevel gears. The spiral driving-gear, G, in the illustration, Fig. 109, is, it will be seen, free to slide up on the driving-sleeve a small distance; excepting for the pressure by which it is held down by the spring S. The object of this arrangement is to give elasticity to the sudden starting of the governor.

Rite's governor is occasionally used for regulating the distribution of the charge in the gas-engine. The valves are then actuated by means of an eccentric, the position of which is under control of the governor.

In connection with large multiple-cylinder engines where several fuel-valves are to be regulated by the governor, its power may become insufficient to handle the regulation, directly. In such cases an indirect regulation may be effected by means of a

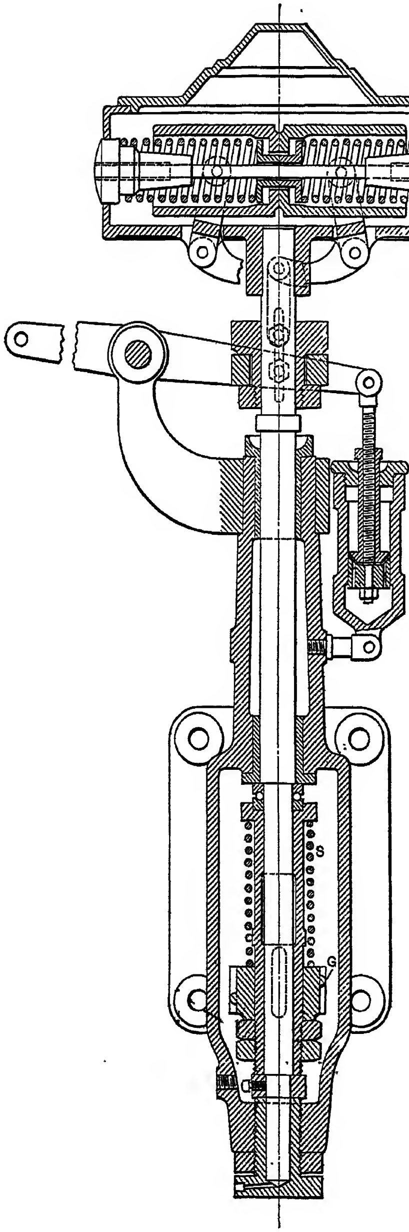


FIG. 109.

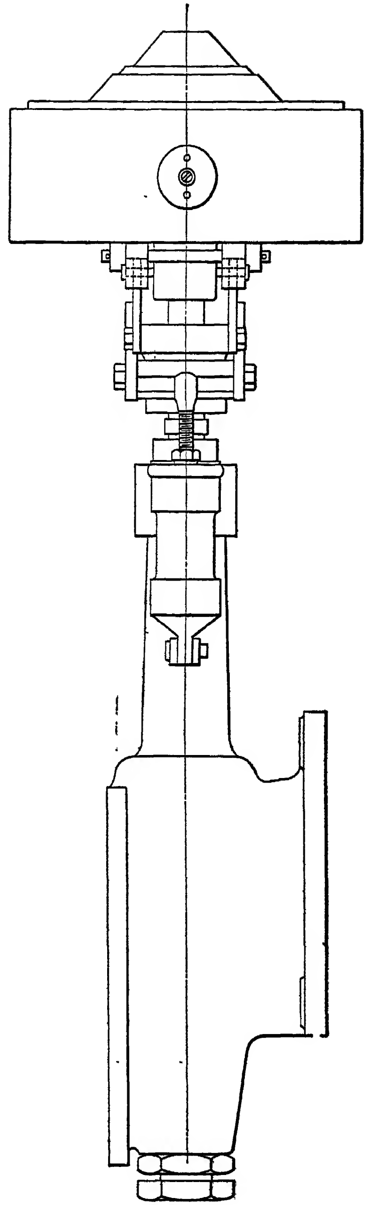


FIG. 110.

hydraulic piston, the motion of which is controlled by the governor. Fig. 111 represents, diagrammatically, an arrangement for such an indirect regulation that has occasionally been used for the governing of large engines.

Referring to the figure, P, P_1 are outside packed plungers actuating the controlling lever, M, which regulates the engine valve-gear. The governor is shown in its low position—the

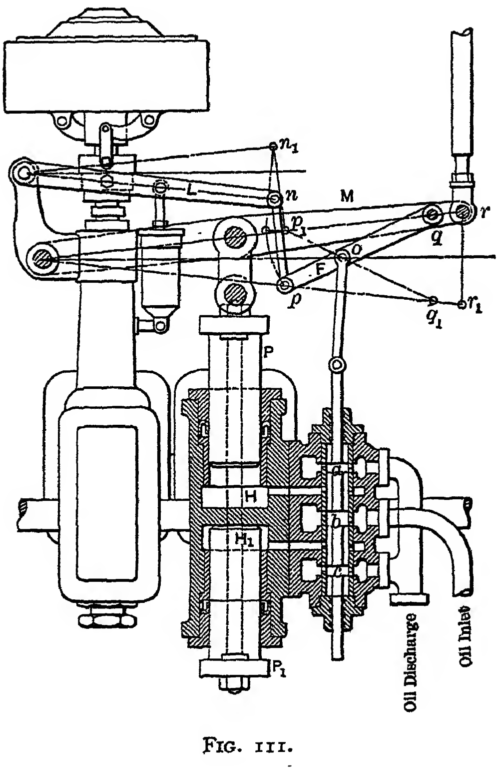


FIG. 111.

governor-lever L being down, and the hydraulic plungers held up by the pressure acting in the top chamber of the hydraulic cylinder.

If the governor rises, say, to its top position, it will lift the floating lever pin p to p_1, and also lift the hydraulic controlling-valve, which consists of the three spools a, b and c having a sliding

fit in the valve-chamber sleeve. The oil, under pressure, will then be admitted below the middle spool, *b*, through small holes drilled in the sleeve, and enter the lower hydraulic chamber, H_1. At the same time there will be effected an outlet for the oil from the top chamber, *H*, when the top spool, *a*, rising, uncovers the outlet port of the controlling-valve. The hydraulic plungers will, thus, descend, bringing the lever *M* down, until the pin *q* comes to q_1. The floating lever will then have the position p_1 o q_1, and will thus have brought the controlling-valve back to its normal position for closing the oil-admission and discharge.

Accordingly, to any position to which the governor-lever *L* is brought, to a corresponding position the hydraulic plungers will instantly bring the lever *M*, and hold it there, until further called upon by the governor to again regulate its position.

Advantages of the Different Regulating Systems.—The factors that determine the output of an engine are: The amount of gas admitted, the amount of air admitted, the compression effected, and the timing of the ignition.

To effect governing, two or more of these features are, as has been seen, generally changed simultaneously.

In the hit-or-miss system the gas alone, or the gas and air, are shut off entirely at excessive speeds, but other features remain unchanged.

In throttling an already completed mixture the gas and air volumes are changed proportionally, and, thus, the quality of the charge remains unchanged, but the compression will be diminished.

By having the gas and air throttle controlled separately the quality of the mixture may be changed, but the quantity unchanged, and, thus, the compression unchanged.

Between these proportions the quality of the charge may be changed to any extent, resulting in a more or less decreased compression. It may even be possible to dilute the charge to such an extent that its quantity and compression become greater at reduced loads.

In Fig. 112 are represented a full-load and a light-load card from a throttling engine.

The initial pressure, the compression, and the mean effective pressures are as follows:

Of the full-load card—

Initial pressure.	$p' =$	14 pounds per square inch.
Compression pressure.	$p'_c =$	104 pounds per square inch.
Mean effective pressure.	$p'_m =$	70 pounds per square inch.

Of the light-load card—

Initial pressure.	$p'' =$	10 pounds per square inch.
Compression pressure.	$p''_c =$	72 pounds per square inch.
Mean effective pressure.	$p''_m =$	20 pounds per square inch.

Assuming V_f and V_l to be the volumes of the mixture admitted per stroke, of standard temperature and pressure, respectively, at full and at light load; and assuming the temperature of the charge to be at both occasions the same, then we have

$$\frac{V_f}{V_l} = \frac{p'}{p''} = \frac{14}{10} = 1.4.$$

This ratio represents also the ratio between the heating-value admitted per stroke at the two instances. The ratio between the work performed during the full load and the light load period is represented by the ratio between the mean effective pressures, and is

$$= \frac{p'_m}{p''_m} = \frac{70}{20} = 3.5.$$

Hence, the efficiency is $\frac{1}{1.4} \times 3.5 = 2\frac{1}{2}$ times as great at full load as at light load.

The main cause of the poor efficiency at light loads is the slow combustion of the charge, due to the low compression. It would, therefore, be desirable to increase the compression at light loads, which cannot be done, however, excepting by diluting the charge with more air; and the question arises, if a quicker combustion and higher efficiency will be obtained from a poorer mixture and higher compression. With respect to some fuel-gases this appears to be the case, and, on this account, a so-called constant-quantity regulation is in many cases the more economical.

Fig. 113 represents a full-load and a light-load card from a constant-quantity regulation, and, although the combustion at

light load is improved upon compared with the condition in Fig. 112, the maximum pressure occurs, even here, too late in the stroke. The suggestion is then near at hand, to ignite the charge earlier

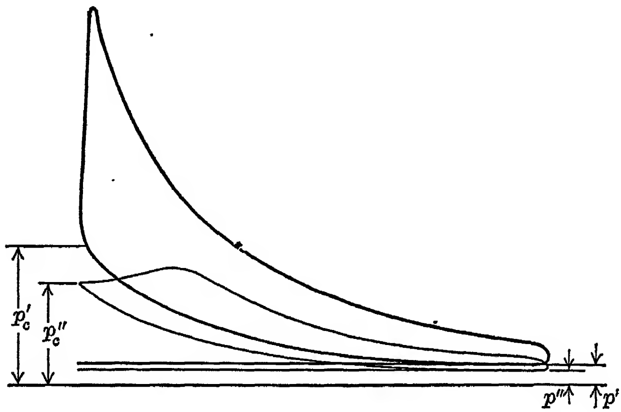


FIG. 112.—Constant-Quality Regulation.

at light load, so as to obtain a more complete combustion at commencement of the pressure-stroke. Regulations of this order, combining a constant-quantity regulation and advancing ignition, have been tried, and the appearance of the cards would indicate

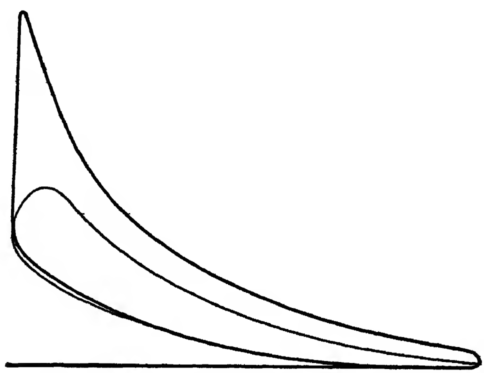


FIG. 113.—Constant-Quantity Regulation.

an improvement upon the simple throttling, or cut-off, governing. Any decided improvement in economy has, however, failed to appear in practice.

The diagram Fig. 114 represents a full- and a light-load card of a constant-quantity regulation, with advancing ignition on light loads. It will be noticed how in the light-load card, represented in fine lines, the ignition takes place early in the stroke, causing the initial pressure to become quite high. The question may, however, be asked, if the increased frictional work due to a high resistance on the compression stroke does not, actually, offset what little advantage there is derived from a more perfect combustion.

It was brought out, in connection with the subject of the balancing of the reciprocating parts, that in heavy engines, particularly in engines of the tandem type, a high compression is of

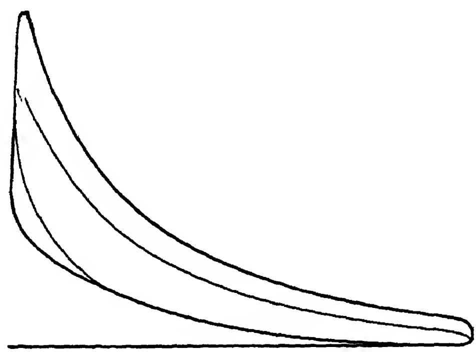


FIG. 114.—Constant-Quantity Regulation with Advancing Ignition.

advantage for quiet running of the engine. It would therefore seem that the constant-quantity regulations, in which the compression remains the same for all loads, would particularly be desirable for heavy engines.

It being a fact that a weak charge requires a higher compression for rapid combustion than a richer one, it has been suggested and tried, to compress, at light loads, a charge of a highly diluted mixture to a higher pressure than the regular charge at heavy loads. In other words, the rich mixture at heavy loads is throttled to the required extent for avoiding pre-ignitions, and the weaker mixture is admitted at a greater density to obtain

a high compression. In the diagram Fig. 115 are shown a full- and a light-load card representing a regulation of this kind,* in combination with an advancing-spark regulation.

Theoretically, the combination regulating systems appear to be ideal, but, practically, the advantage gained in a higher efficiency is often offset by the greater complications they involve. There are, however, cases when the nature of the fuel, and the load-conditions, are such that an improved combustion is of importance, disregarding whatever improvement in efficiency may be effected.

In throttling engines drawing the fuel-mixture from a mixing-chamber, particularly when the latter serves two or more cylinders, there is often, at light loads, difficulty from back-firing into the

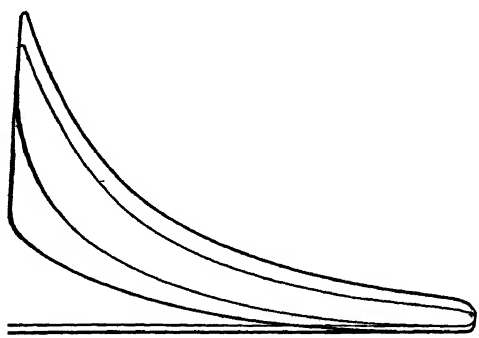


FIG. 115.—Regulation with Increasing Quantity at Light Loads.

mixture. It is due to slow combustion of the fuel when insufficiently compressed; the charge still holding fire when the inlet valve opens for a following suction-stroke. With fuels not readily vaporized, or poorly mixed, difficulties of this nature will be serious. In such cases a high compression and an early ignition will be very essential, and the combination governing system may be used to good advantage.

Factors that Enter the Problem of the Governing of Gas-

*This regulation was a feature of the Letombe gas-engine built by the Société Anonyme d'exploitation brevets Letombe at Lille, and exhibited at the Paris Exposition.

Engines.—The conditions under which the gas-engine operates make the perfect governing of the engine much harder to accomplish than is the case with steam-engines. The disturbances that militate against a good regulation of the speed, disregarding those due to explosion waves about which not enough is known to tell exactly what influence they may have on the regulation, are, principally, pre-ignitions, back-firing, and fluctuations in the explosion-pressure.

Pre-ignitions and back-firing, which may be considered as being of only temporary nature, are readily guarded against when the cause of the disturbances is understood; and after an engine is once adjusted to the conditions under which it is working, such disturbances are, generally, practically eliminated.

Pre-ignitions, or self-ignitions, are often due to one or other of the following causes:

Excessive compression of the fuel used; an uneven gas-mixture, a too readily ignitible rich charge being occasionally supplied; an over-heated exhaust-valve, piston, ignitor, or some protruding and poorly jacketed part of the combustion-chamber; a poor grade of cylinder oil allowed to leak past the piston and becoming carbonized in the combustion-chamber; foreign matter and impurities admitted with the gas or air charge, collecting in the combustion-chamber and holding the fire; a too rich fuel-charge which, due to incomplete combustion, leaves a carbon deposit in the cylinder.

If due to the over-heating of some part in the combustion-chamber, a more thorough water cooling, or the removal of some unnecessarily protruding part may become necessary to stop pre-ignitions; or, occasionally, the reduction of the engine-speed may help matters.

The usual causes of back-firing into the mixing-chamber are, in general: slow combustion due to a low compression, a fuel poorly vaporized, or leaky valves. The effect from back-firing may be very much reduced, or entirely eliminated, by doing away with the mixing-chamber, and effect the mixing of the fuel and air first at the inlet-valve. Should the inlet-valve leak, it would, of course, with this arrangement still be possible that the flame

might throw back into the gas and air pipes, but any serious disturbance will not be caused.

In Fig. 116 is shown a continuous diagram of the explosion-pressures obtained for a series of explosions. Diagrams of this description are readily obtained by advancing the paper-drum of the indicator, steadily, for a series of explosions, while the pencil-arm records the height to which it is driven by the explosion-pressures. The diagram illustrated, which was taken during a

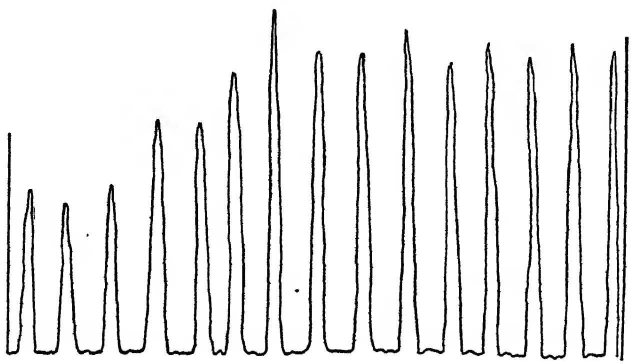


Fig. 116.—Continuous-Explosion Diagram.

period of partially light and partially heavy load, but, at each period of perfectly steady load, shows the fluctuations in the maximum pressure from cycle to cycle.

The causes of the variation in the explosion-pressure, under constant load-conditions, are of a variety of kind, and cannot readily be guarded against.

The more evident causes of the fluctuating pressures are:

An uneven mixture, due to pulsations in the gas, the atmosphere, or the exhaust; an uneven compression, due to an unsteady governor, due to pulsations in the mixture supply, or due to poorly guided valves, causing the valves to seat themselves more or less tight; variations in the point of the ignition, probably due to the springing of, or due to the inertia of, the ignition-device, causing the spark to be timed unevenly.

With a reasonably heavy fly-wheel, the effect of the fluctuations

in explosion-pressure is not generally serious enough to impair the steadiness of the speed of the engine, but, in order to obtain the most effective governing, the effort should be to eliminate as many of the enumerated disturbing influences as possible, without the introduction of complications of detail, that, on the other hand, will in themselves become objectionable.

CHAPTER XIII

ENGINE AUXILIARIES

Carbureters.—When liquid fuels are used, a fuel-vaporizer, or what is commonly called a carbureter, becomes a necessary auxiliary to the gas-engine. The three most generally employed modern types of carbureters for gasoline are shown in Figs. 117, 118, and 119. The one illustrated in Fig. 117 is of the type in which the gasoline is supplied to the vaporizer under a slight head; a check-valve being utilized to check the flow during the intervals between the suction-strokes. The check-valve is, in the apparatus illustrated, made large, serving also to check the return flow of the carbureted air. This type is often used in connection with two-cycle engines, in which the carbureted air is under pressure during the forward stroke of the piston. The gasoline-connection is made at *I*, and the supply is adjusted, as to quantity, by means of the needle-valve, *N*, which may be set by trial to give a suitable opening for the admission of the fuel.

Carbureters of this type are simple, but they are liable, at varying loads and when the gasoline-pipe is of some length, to give trouble. There being no storage of gasoline near the spray opening, pulsations in the gasoline supply-pipe affect the tillflow of the fuel differently at different periods. This difficulty may be overcome by fitting in the fuel-pipe near the carbureter a constant-level reservoir from which the fuel will be drawn directly into the supply-opening to the carbureter.

It must be borne in mind that, in order that the gasoline-engine shall operate properly, the fuel required for the proper mixture at any load-condition must be supplied uniformly, and that the engine will never act well if there is the slightest cause for sudden and extreme changes in the composition of the mixture, which are not called for by any variation in the load. The more sensitive the governor is, other conditions being equal, the more sensitive the engine will be for pulsations in the fuel-supply.

Assuming that, due to disturbances in the fuel-supply, a heavy charge be admitted at a period of light load, it is evident then that a following charge would, through the action of a sensitive throttling governor, become extremely weak, and apt to cause back-fire into the mixing-chamber due to slow combustion. Again, alternate very rich charges are apt to cause pre-ignitions due to an over-supply of the fuel.

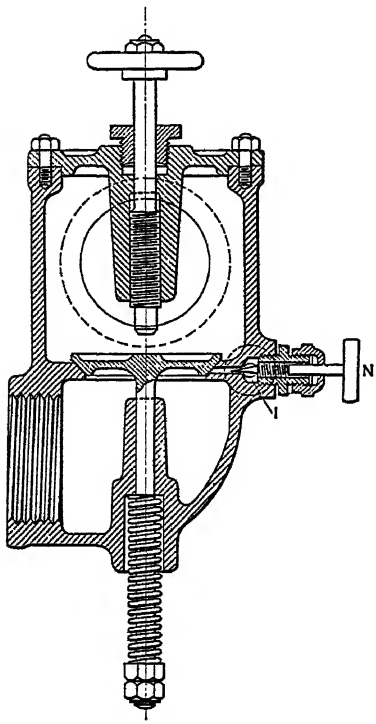


FIG. 117.

Fig. 118 is a constant-level reservoir-carbureter. In connection with this carbureter there is a small gasoline pump driven by the engine, which supplies a suitable amount of the fuel from the fuel-tank into the reservoir *R*; keeping it constantly filled to a certain level, one-half to one inch below the spray-nozzle *N*. Whatever amount of fuel the pump supplies above what is used by the engine will be returned through the overflow pipe *O* to the fuel-tank. At each suction-stroke of the engine, gasoline will be drawn into the vaporizer-chamber, due to the slight vacuum created there, and it will be absorbed by the air entering through the air-supply pipe *A*. Before starting the engine, the fuel must, of course, be pumped by hand into the reservoir, to supply the requirements for the first explosions.

This carbureter is often used to advantage in connection with stationary four-cycle engines.

Fig. 119 is a type of float-feed carbureter, which is most generally employed for all classes of gasoline engines. *C* is a small float-chamber, in which the fuel is held at a constant level

by means of the float, *F*, and inlet valve, *V*, operated by it. From the float-chamber the fuel is supplied to the nozzle, *N*, through which it is sprayed during the suction-strokes into, and absorbed by, the passing air-current entering through the air-supply port *A*.

It is often the case that the carbureted air leaving the carbu-

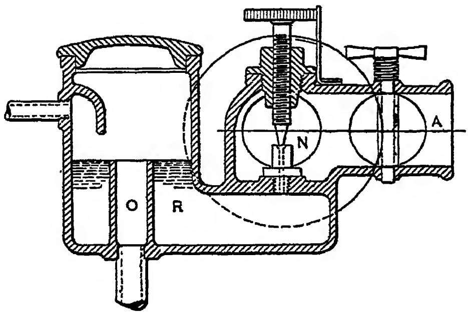


FIG. 118.

reter is too rich in gasoline-vapor for complete combustion, and it becomes necessary to mix it with additional air. The secondary air-supply is, in the apparatus illustrated, taken in through the orifice at *O*, and it may be throttled to any extent required, by

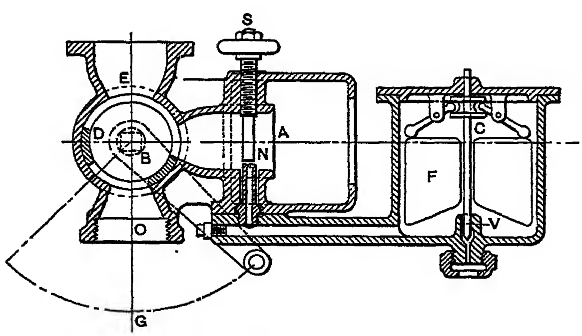


FIG. 119.

bringing the throttle-valve handle more and more over toward the position *B G*. After this position is reached the cut-off valve-bridge, *D*, comes into action, closing the inlet, *E*, to the cylinder.

There is no needle-valve used for regulating the orifice of the

spray-nozzle; the latter being made permanently of a size to correspond to the requirements of the engine. The screw-spindle *S* allows, however, to some extent, an adjustment to be made.

The by-passing of air through the secondary air-inlet is necessary in engines of a considerably varying speed, in order to overcome the tendency of the suction spray-nozzle to supply an excess of fuel at high piston-speed, when the vacuum in the mixing chamber becomes high.

The carburetion of gasoline will be more effective when dry and slightly heated air is used for absorbing the fuel-vapor; and the air-supply is, therefore, often drawn from a place where the heat from the engine-cylinder or exhaust-pipe will have for effect to pre-heat it to some extent. The pre-heating of the primary air-supply will, particularly, be desirable when, as often is the case, only part of the final air-charge, one-third or less, is expected to vaporize the full quota of gasoline.

Alcohol Carbureters.—The same carbureters which are used for gasoline may, with certain restrictions, also be used for vaporizing alcohol-fuels. For alcohol the pre-heating of the air is much more necessary than with respect to gasoline-fuel, because the former fuel vaporizes much more slowly. The object should also be to minimize the vacuum in the vaporization-chamber, because a high vacuum tends to chill the air and thus prevent it from readily absorbing the fuel. On this account, the port-openings through the apparatus, for both air and fuel, are required to be more ample than in the case of some more readily vaporized fuels. The entire air-supply should pass over the spray-nozzle, or, at most, only an unimportant part of the supply be by-passed.

Principal Auxiliaries of the Automobile Motor.—In Fig. 120 is represented a modern automobile engine, in connection with its fuel-supply system and cooling system. The former consists of the gasoline tank, from which the fuel is supplied to the float-case and from there to the carbureter; always at a constant head. In the carbureter a fuel throttle is applied, by means of which the speed and power of the engine can be regulated by the driver;

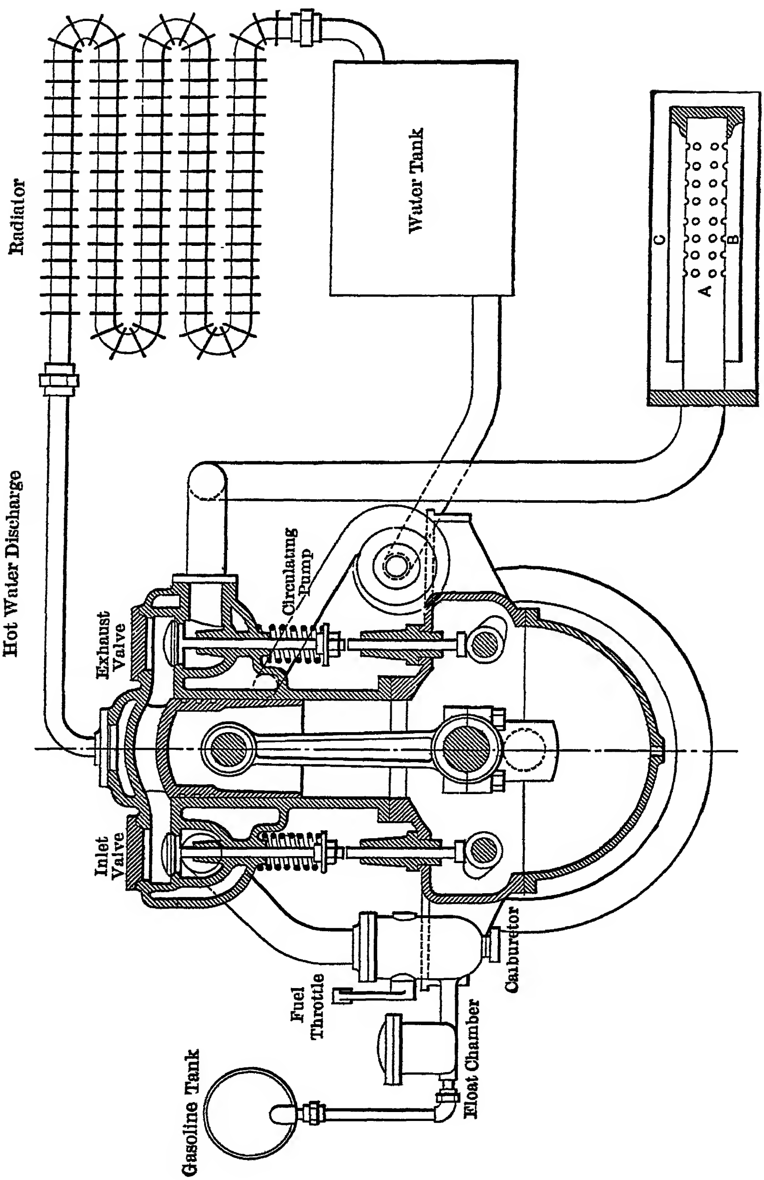


FIG. 120.—Automobile Engine and Auxiliaries.

or the fuel may be cut off to the extent of merely keeping the motor, empty, in motion. The gasoline tank is, in practice, not always located so as to feed by gravity, as in the figure, but a slight air-pressure may, by means of a hand-pump, be created in the tank, by which the fuel is forced out, and in to the float case, with equal certainty wherever the tank may be located.

High-power machines must always be cooled by means of a water-cooling system, since the cooling-surface required for the dissipation, directly to the air, of the heat that necessarily must be carried off is, in these, far in excess of that which the engine itself can well present. A radiator composed of a number of ribbed pipes, can however, always be built to present any surface required; and when in motion, the air then being set in a rapid circulation through its coils, such an apparatus is very effective for dissipating heat.

A small rotary pump is used to effect a good circulation in the cooling-system. The hot water passing off from the highest point of the jacket is carried through the radiator, in which its temperature is rapidly lowered and after having been cooled it collects at the bottom of the water-tank below, from which it is again drawn by the pump and forced in at the lower part of the engine jacket.

The Exhaust Muffler.—An exhaust muffler, or silencer, is shown connected to the discharge-pipe from the engine in Fig. 120.

In order to prevent the noise incidental to the escape into the atmosphere of the exhaust from the gas-engine, the remaining heat-energy of the gas, its capacity for doing work, which will be transferred into noise at its escape, must be dissipated.

Its pressure might be reduced, or its volume increased, by which two alternatives its sensible heat becomes reduced. Further its potential energy must be dissipated by reducing the velocity of the escaping gas.

These requirements are effected by changing, gradually, through expansion, the heat-energy of the discharge into velocity of the gases, and by reducing this velocity, by means of baffles, to a desirable limit, before the gases are allowed to escape. Of

course a direct cooling of the gas in the exhaust pipe, if practical, will greatly help to reduce its heat-energy and to silence the exhaust.

In the apparatus illustrated, the gradual expansion of the gas is effected by allowing it to escape through numerous holes shown in the pipe *A*. The velocity acquired by the various streams of gas when expanding in these nozzles is then readily baffled in the chamber *B*, and still further in the return-chamber *C*.

The muffler, unless it is made quite large, always throws some back-pressure on the engine, and it is on this account not a very desirable apparatus, but it is, in many cases, a necessary auxiliary to the gas-engine.

Ignition Devices.—The modern method for igniting the charge in the gas-engine cylinder, excepting in certain classes of kerosene- or oil-engines which are self-igniting, is by means of an electric spark. The requisite current may be obtained in various ways. By means of a common cell-battery, by a storage-battery, from an electric service-circuit, or by means of a small special dynamo or induction-magneto.

The electric spark is formed in two ways. By the so-called jump-spark system, and by the make-and-break system.

In the former system there is provided, in the secondary circuit from an induction-coil, a gap between two sparking-points in the cylinder. When thus the primary circuit, in series with the battery, is opened or closed a spark will be formed between the sparking-points—it, so to speak, jumping across the gap between the points. Hence the name of the system.

Fig. 121 illustrates the general scheme for the jump-spark system. *I* is the induction-coil, *B* is the battery, *V* the wiper opening or closing the primary circuit by coming in contact or out of contact with the contact-piece *C*. *S* represents a standard spark-plug screwed into the head of the cylinder and connected at its terminals with the leads from the secondary winding of the induction-coil.

With the wiper *V* touching the contact *C*, the primary circuit may be opened and closed a series of times by means of an ordinary vibrator, such as is used on a *Rhumcorf* coil, in which

case a number of sparks are formed at the sparking-points. Ordinarily, however, the vibrator may be left out and then two sparks only will be obtained, one when the wiper *V* closes and one when it opens the primary circuit.

The Make-and-Break System.—In the jump-spark system the current must be of high tension to cause a spark to form between the sparking-points; these being generally set $\frac{1}{32}$ to $\frac{1}{16}$ of one inch apart. In the make-and-break system the sparking-points are in contact before the time for a spark. When the circuit then is

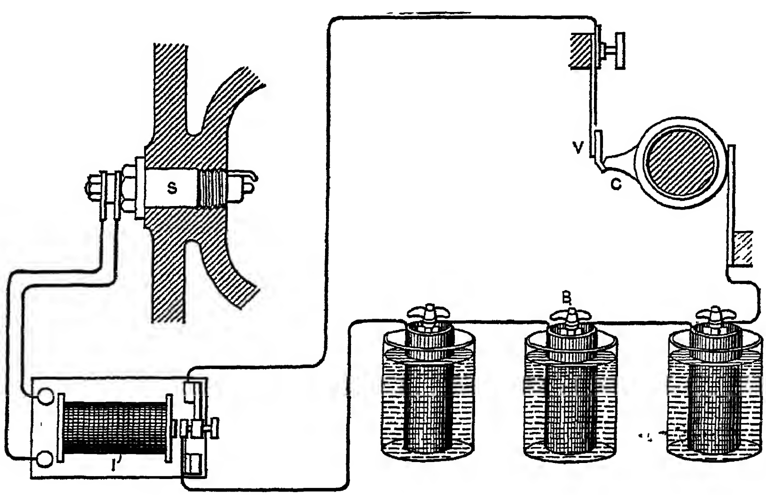


FIG. 121.

broken, suddenly, by moving apart the contact-points, a good spark will be formed with a current of less tension than that required in the jump-spark system, and the deterioration of the sparking-points will be less. The primary wire of an induction-coil is often inserted in the circuit, as in Fig. 122, in order to intensify the spark.

Fig. 122 represents the arrangement of an ignition device of the make-and-break system. In the circuit from the battery, *B*, is inserted the induction coil *I* and the leads, l l_1, connecting with the fixed and movable electrode. Ordinarily, the circuit is open by having the movable sparking-point thrown back in the position

a by means of the coil-spring *S*. *C* is a crank secured to the end of the valve-gear shaft, and it carries one end of the pick-blade lever, *L*, whose other end is secured by the pin *d* to the link, *H*. By the same pin is also secured the pick-blade, *P*, which is also held in position by a stiff spring *T*. When the shaft revolves in the direction of the arrow, the pick-blade will, at the proper time, just before firing, engage with the lever *M* and swing it toward the right, thereby closing the electric circuit; the spring *T* serving to press the contact-points firmly together. At the time for firing

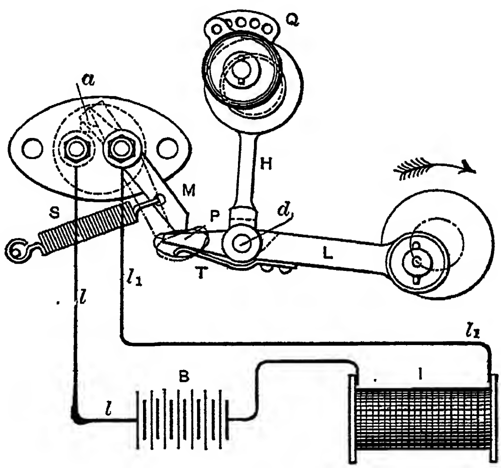


FIG. 122.

the lever *M* will disengage and suddenly break the contact between the sparking-points, causing a spark to be formed.

Instead of the battery, a small generator driven by the engine may, of course, be used, and in such a case the battery, *B*, and inductive-coil, *I*, would be switched in the circuit only for starting the engine.

The inertia of the make-and-break mechanism tending, at high speeds, to make the timing of the spark more or less uncertain, it is apparent that the jump-spark system will be the more suitable for engines of a high number of revolutions.

Magneto Ignition.—The latest and simplest device for generating a current for ignition is by means of the induction-magneto.

The *Bosch* type of magneto, which is now often found to work in connection with all classes of engines, consists simply of a small Siemens armature which is made to oscillate between the poles of a permanent steel-magnet. Fig. 123 shows the complete arrangement of the ignition device in connection with this apparatus. *M* is the magneto. On the armature-shaft is secured a *T*-crank which is pulled positively to its neutral position by the springs *S S*. *C* is the cam-shaft of the engine. On the end of it there is secured the crank *F* carrying one end of the pick-blade lever *L*. When the shaft revolves in the direction of the arrow, the pick-

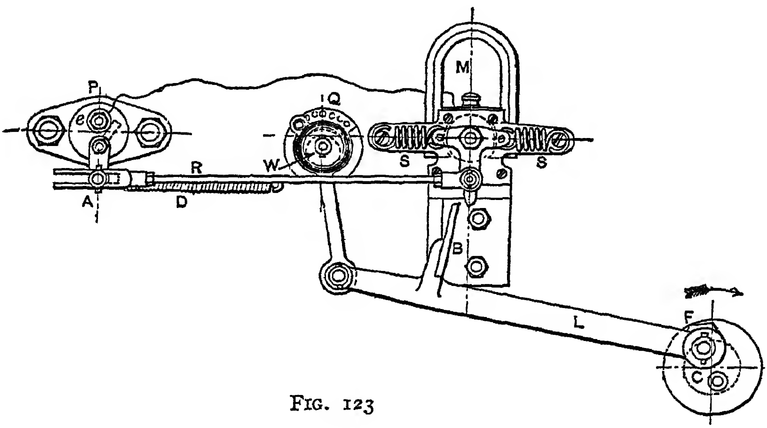


Fig. 123

blade *B* will, at its motion toward the right, engage with the *T*-crank, carrying it out of plumb toward the right. At the proper moment for firing, the *T*-crank is disengaged and is pulled quickly back by the springs *S*. A current is thereby generated in the windings of the armature, which is led to the stationary electrode *e* of the spark-plug *P*. At the moment when the *T*-crank is disengaged and swings to its normal position, the forked end of the push-rod *R* will give to the crank-arm *A* of the movable electrode a blow which causes it to swing to the left, thereby momentarily breaking the contact between the sparking-points in the cylinder. These points are normally held in contact by means of a weak coil-spring *D*, which is in the figure partly covered by the push-rod. In returning to the left the pick-blade

clears the end of the *T*-crank, and by lowering or raising the left-hand end of the pick-blade lever, by means of the link, eccentric and hand-wheel *W*, the release of the *T*-crank can be made to occur earlier or later, according to the desire to advance or retard the spark. The timing of the spark is thus accomplished by turning the hand-wheel *W* to the right or left and locking it in the various holes of the quadrant *Q*. The timing-device of the igniter, Fig. 122, is very much of the same arrangement.

In some makes of engines the timing of the spark is accomplished automatically by the governor; the object being to effect

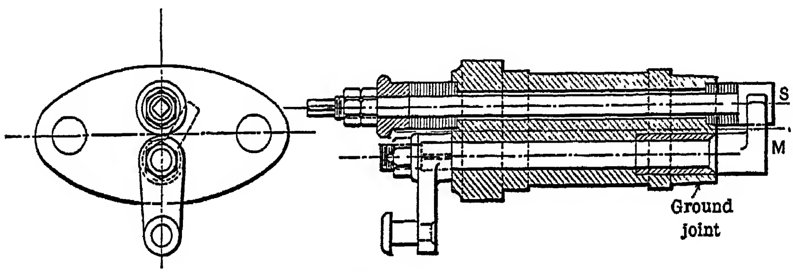


FIG. 124.

an earlier ignition at light loads, when the combustion is slow, due to a lean mixture and low compression.

In Fig. 124 is shown the general construction of a spark-plug for the make-and-break system. The stationary electrode *S* must be carefully insulated, electrically, by means of mica or lava washers and bushings, whereas the movable electrode *M* stands in electrical contact with the housing. There is, therefore, only one lead required from the magneto to the stationary electrode as shown in Fig. 123.

Similarly, the lead $l_1\ l_1$, in Fig. 122, may, if more convenient, be connected to any point of the engine which stands in electrical contact with the housing of the spark-plug of a construction as described.

CHAPTER XIV

VARIOUS ENGINE-TYPES

The Two-Cycle Engine.—The two-stroke cycle, or the two-cycle engine, is suitable, mostly, when the main object is to obtain the greatest simplicity in a small engine, or the greatest amount of power by the smallest weight in a large engine.

Figs. 125 and 126 show the main features of the two-cycle engine in its simplest form, and its action is as follows: During the up-stroke of the piston, as in Fig. 125, the charge of suitable mixture is drawn in to the closed crank-case *C* through the opening *A*, and it becomes compressed, slightly, during the down-stroke—to about 6 to 8 pounds above the atmosphere. When the piston, passing down, comes to a proper position near the end of the stroke, the exhaust port *E* opens first for the exhaust gases to pass out from the pressure side of the cylinder, and, later, the inlet port *I* will begin to open for new charge. Both ports become fully open when the piston arrives at the lower centre, as shown in Fig. 126. The charge will enter from the crank-case in to the cylinder, expelling the neutrals before it, until, soon after the beginning of the upward stroke, the inlet port first and then the exhaust port become closed by being covered by the piston, and the compression of the charge commences. When fully compressed, at the time the piston arrives at the top position, the charge will be ignited, and the power stroke will follow during the down-stroke.

Thus, every down-stroke will be a power-stroke and every up-stroke a suction-stroke. Two full strokes of the piston constitute one cycle.

A closer regulation of the charge and a more reliable action can be obtained by using an inlet-valve with adjustable spring-tension. Figs. 127 and 128 illustrate a small marine engine with such an inlet-valve, of the design of Smalley Bros. Co., Bay City, Mich. The speed-regulation of engines of this type is sometimes

effected by the thrŏttling of the charge in the inlet-valve, in which case means are provided for preventing the inlet-valve from lifting to its full lift, when a reduced speed is required.

The compression of the charge, instead of being performed by the working piston, may, of course, be accomplished by a special

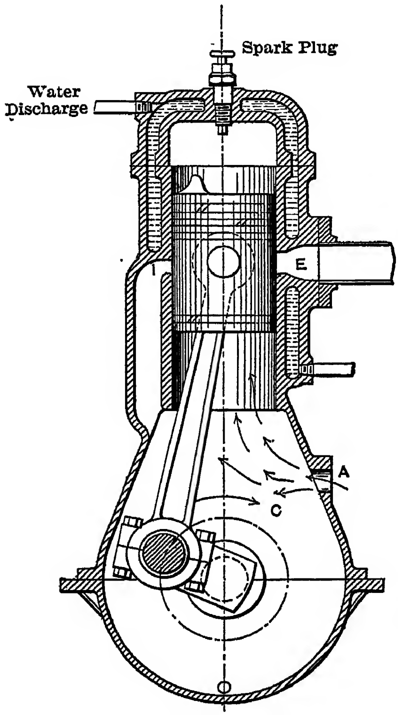


FIG. 125.

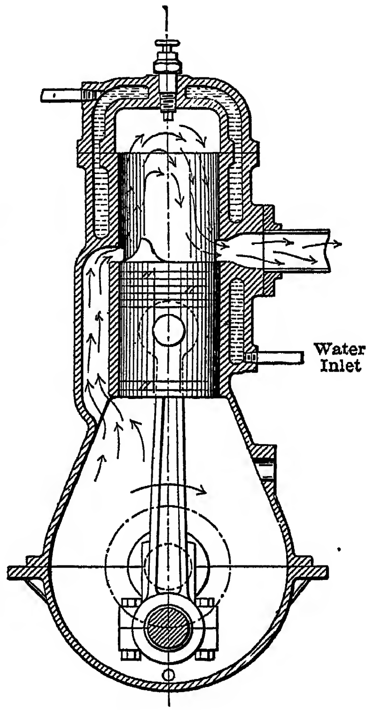


FIG. 126.

compressor-cylinder, and in large two-cycle engines this will be nearer at hand.

The Koerting Two-Cycle Engine.—The Koerting double-acting, two-cycle engine is a type in which the air and gas for the charge are compressed and delivered separately by auxiliary pumps driven by the main engine shaft. The pump-pistons are so proportioned and their motion so timed with respect to the main working piston that the proper amount of air and gas will be delivered to the working cylinder at the proper time. It is

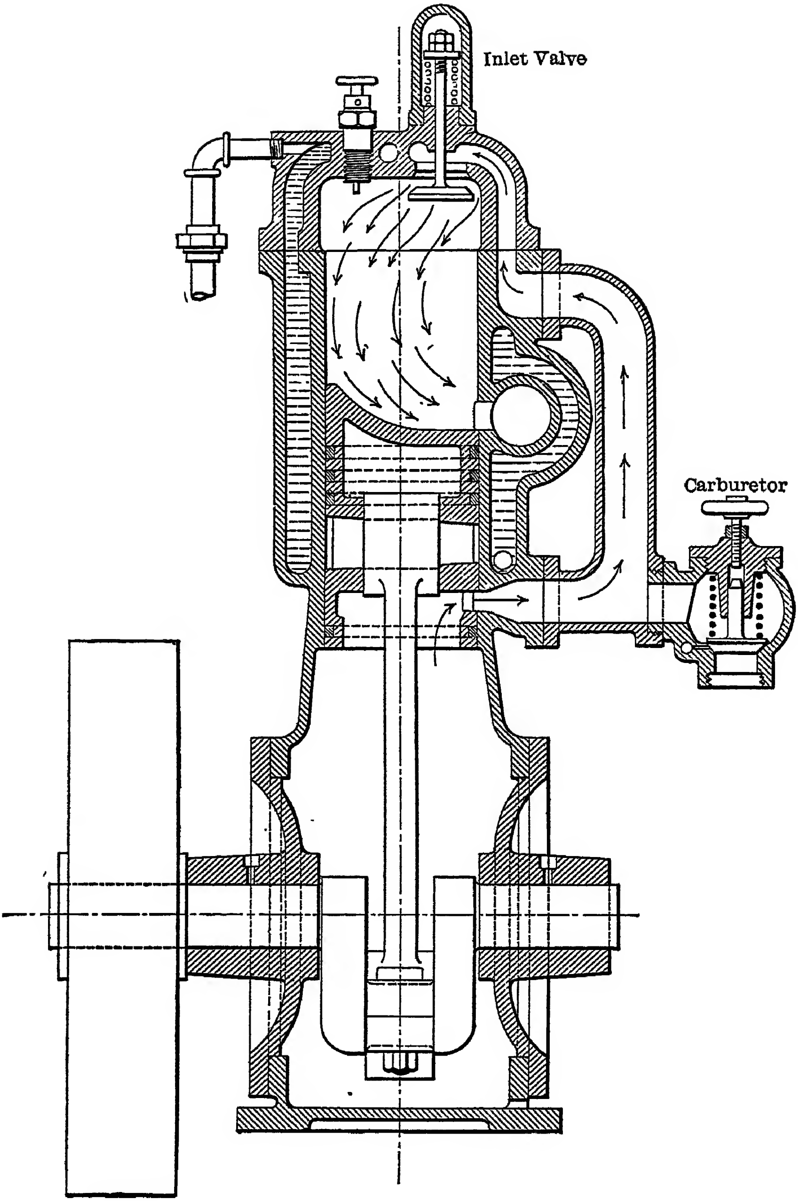


FIG. 127.

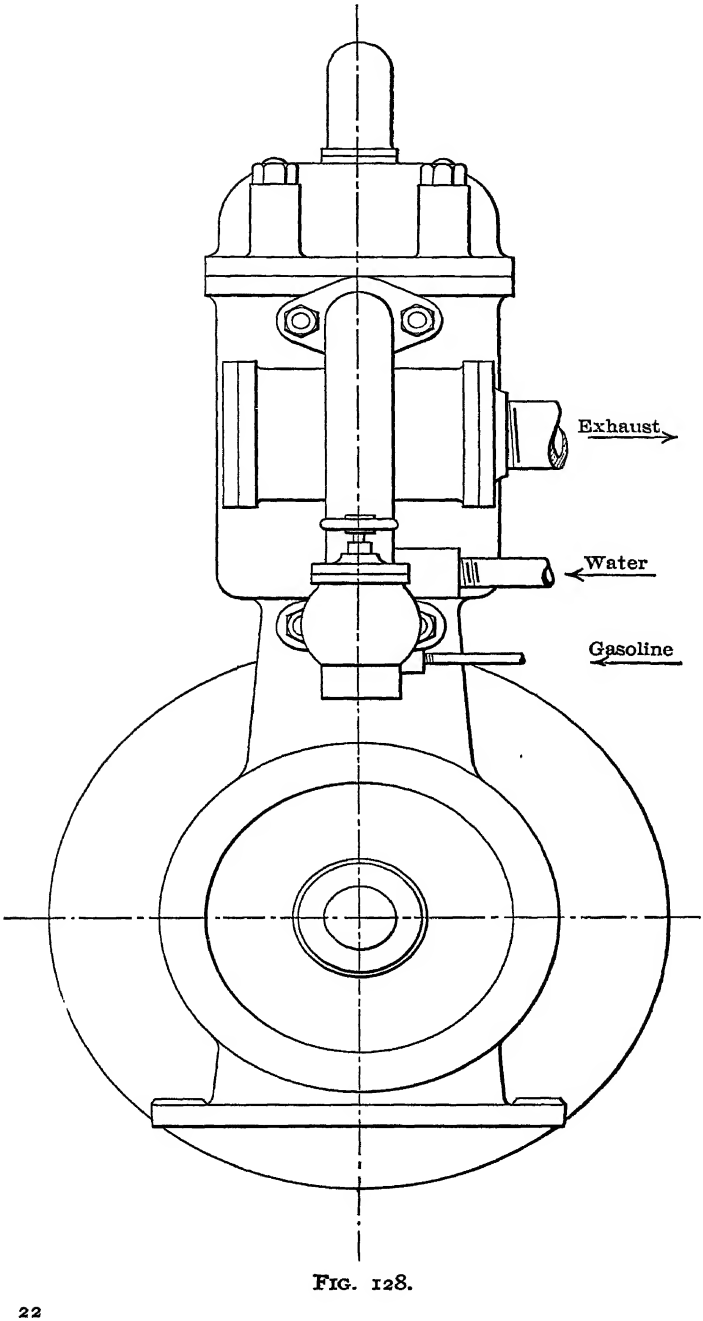


FIG. 128.

to be noted that the gases remaining in the cylinder from a previous expansion-stroke are not, in the two-cycle engine, removed by a following exhaust-stroke, and it becomes therefore necessary to resort to scavenging, in order to remove the neutrals which fill the cylinder after the expansion-stroke is completed. This is in the Koerting engine, as in two-cycle engines generally, accomplished by forcing a current of air through the cylinder while the exhaust port remains opened, and it becomes the function of the air pump to furnish also the necessary air for the purging of the cylinder.

Figs. 129*a* and 129*b* are a plan and a front view which show

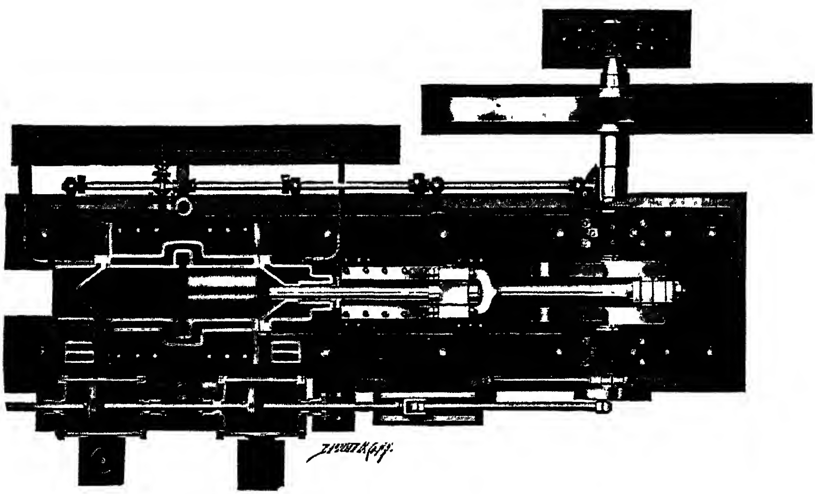


FIG. 129*a*.—Koerting Two-Cycle Engine. Sectional Plan.

the general arrangement of the Koerting engine, and in Fig. 130 the working cylinder and the pumps are shown diagrammatically.

As seen in Fig. 130, there is, for each end of the working cylinder, only one valve—the admission valve; the discharge being exhausted through ports, *M*, at the middle of the cylinder, and these are covered by the piston at all times, excepting at the end of each working stroke. The admission valve-casings *A* and *B*, are provided with separate air- and gas-ports terminating directly above the admission valves.

Referring to the compressor pumps: C is the air pump and D the gas pump, and their pistons are driven by a continuous rod from a crank on the main engine-shaft.

The action of both pumps is controlled by piston-valves—E the air valve and F the gas valve—driven by means of rockers and separate eccentrics from the main engine-shaft (an earlier type of valve gear which is shown in Fig. 129*a* used, however, one eccentric only for both valves). The admission to the air pump is at G, and the air discharge ports are at H H, while the admission to the gas pump is at I and the gas-discharge to the working cylinder at K K. By the paths of the arrows leading

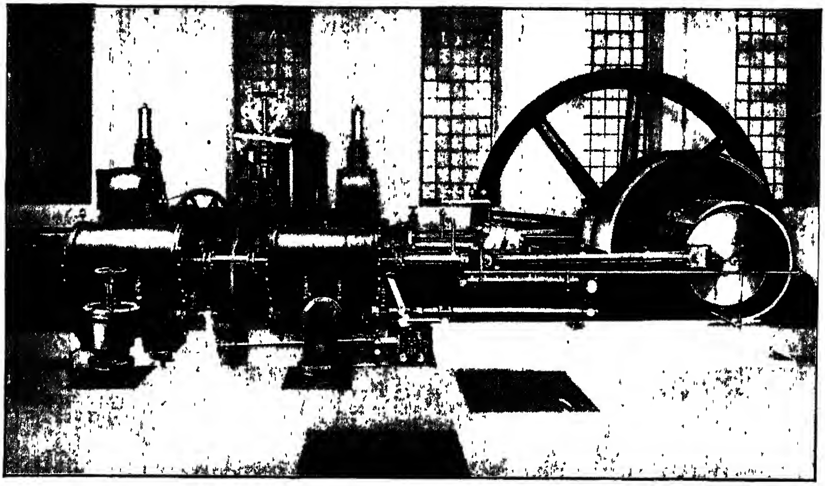


FIG. 129*b*.—Koerting Two-Cycle Engine. Front View.

from the discharge ports of the pumps are indicated the connecting channels between the pumps and the working cylinder, and it will be seen that the crank-ends of both air- and gas-pumps discharge through separate channels to the crank-end of the working cylinder, while the head-ends of both pumps discharge, also through separate channels, to the head-end of the working cylinder; x x are regulating valves to which reference will be made presently.

In Fig. 130 the working piston, P, is represented as having

arrived at the end of its forward stroke, and the exhaust ports are fully uncovered for the discharge to escape from behind the piston. When the piston occupies this position the pressure back of it, in the head-end of the cylinder, has become nearly fully relieved, and the head-end inlet valve, *B*, has opened a small amount. The crank driving the gas and air pumps being

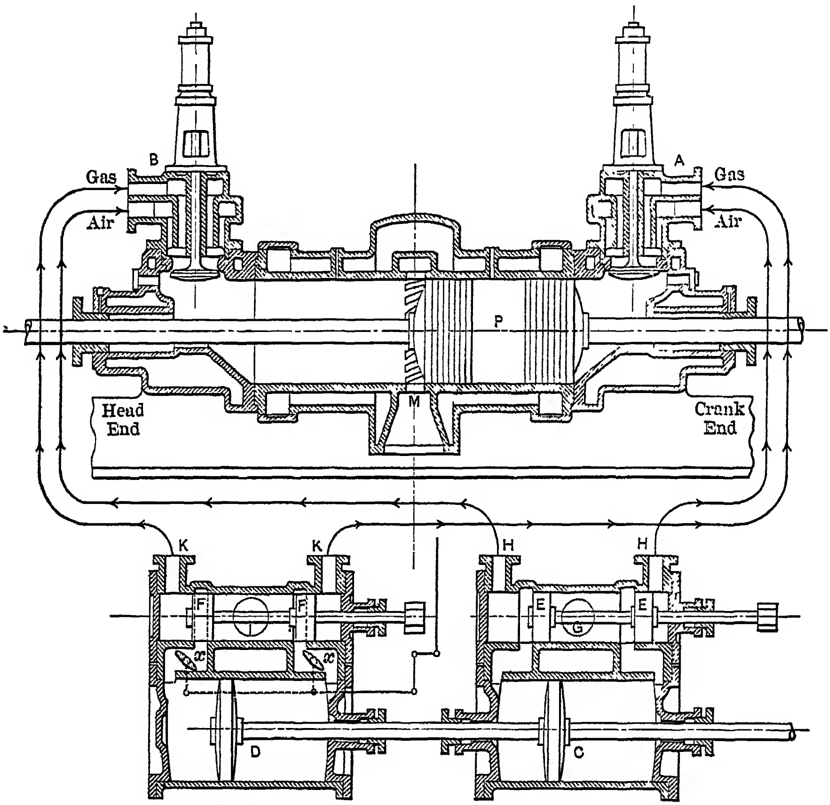


FIG. 130.—Koerting Two-Cycle Engine.

set ahead of the main crank about 110 degrees, the pump-pistons, when the main crank passes the forward centre, will occupy, approximately, the positions in which they are shown in the figure. The piston-valve for the air pump will be open for discharge at the head-end, and the pressure in front of the air piston

will be from 4 to 8 pounds gauge. When, therefore, the main inlet valve opens there will be an inrush of air from the air passage, which drives the burned gases, as completely as possible, out through the exhaust port and fills the cylinder with pure air. The piston-valve controlling the gas pump is, as yet, closed, but it will open immediately for supplying the fuel for the next charge

In the diagram, Fig. 131, the main crank and the pump crank are represented in positions corresponding to those of the working piston and pump pistons in Fig. 130. When the main crank passes the point *a* the exhaust valve will begin to open and it becomes fully open at the end of the stroke, at *c*. At *b* the main inlet valve begins to open, admitting the scavenging air-charge, as explained above, and at *d* the piston-valve for the discharge of the fuel from the gas pump to the engine-cylinder will uncover. This will occur, accordingly, a little later than the position in which the pistons are shown in Fig. 130. Both pump-pistons moving together, they will, from the point *d*, force air and fuel in to the working cylinder, in a proportion that depends on the relative size of the pistons and on the relative resistance the air and the gas will meet in the passages from the pumps to the working cylinder, and they will continue to force the charge in to the cylinder until the main exhaust-ports close, at the point *e* of the diagram. From that point, until the main inlet valve closes, at *f*, there will occur compression of the charge between the pump-pistons moving one way and the working piston moving in the opposite direction. At the point *f*, which occurs approximately when one-third of the charging stroke has been swept through by the working piston, the pressure in the cylinder will be from 8 to 10 pounds, and from there, until the end of the charging stroke, the charge will be compressed by the working piston alone. When at the end of the stroke the charge is fully compressed ignition takes place.

The capacity of the air pump is determined to suit the volume of air required for scavenging and for charge, and this quantity remains the same for all load-conditions. The air-pump piston-valve is set so as to give the maximum efficiency at the compression;

admitting air to the pump cylinder soon after the beginning of the suction-stroke and opening for discharge as nearly as possible when the compression-pressure equals that in the delivery pipe. The gas valve, on the other hand, is set to deliver to the working cylinder a fuel-charge of the required pressure and at the correct moment, which is when the main crank passes the point *d*, Fig. 131, or when the pump-crank stands approximately 120 degrees

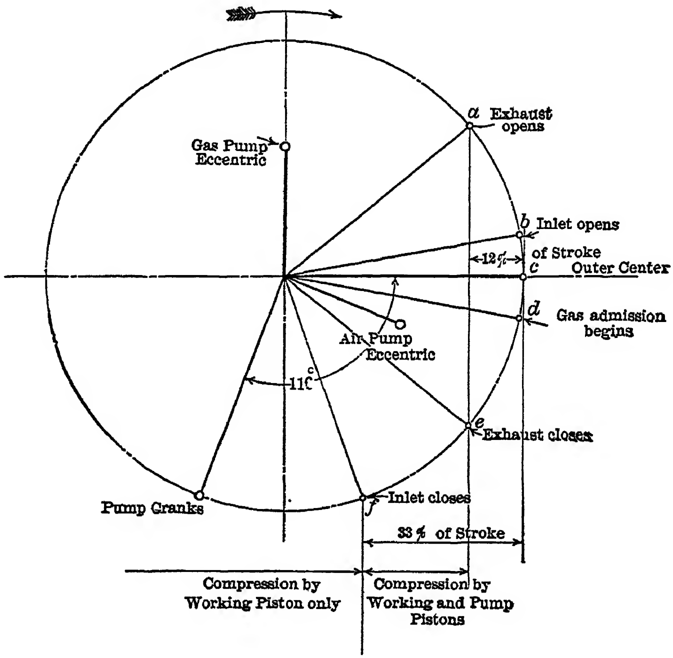


FIG. 131.

from the crank-end centre, *c*. The gas pump, therefore, will not begin to compress its charge at the beginning of the compression-stroke, but will return through the suction-port part of the charge which it has taken in, until the admission-port becomes closed at a point suitable for bringing the compression-pressure to that required at the point *d*.

Accordingly, with piston valves discharging at the outside edges as shown in Fig. 130, the adjustment of the eccentrics for

the air and gas pumps relatively to the crank will be, approximately, as shown by the diagram, Fig. 131.

The Koerting engine is governed on the principle of admitting a variable quantity of fuel in a constant quantity of charge, and this is carried out by supplying the fuel to the working cylinder at a pressure which varies in accordance with the load. The function the governor performs is to regulate, according to the momentary requirements, either the quantity of gas which is taken in to the gas pump for compression or the volume of compressed gas when it is being delivered to the working cylinder; either one or both.

The former may be done by means of a combination inlet and cut-off valve, or possibly by means of a shifting eccentric. The latter simply by throttling and by-passing the compressed gas when it is being delivered to the cylinder.

In the diagrammatical drawing of the Koerting engine, Fig. 130, the latter regulation is represented by the butterfly-valves x x which are assumed to be under the control of the governor.

In the Koerting engine the piston travels 12 per cent of the stroke past the opening edge of the exhaust port, and during the time when the piston travels this distance, back and forth, until the closing of the exhaust port, which is just about one-fourth of the time for one revolution, the three functions, of discharging the waste gases, of scavenging, and of introducing the new charge, must be performed. An engine running 100 turns per minute would, thus, have only 0.15 of one second for performing these functions. It can readily be perceived, therefore, that the velocity with which the gases must be introduced in order to fill the cylinder in this short space of time must be considerable, and this circumstance is known to be the cause of a material loss through fluid friction, which may put a limit to the number of turns the engine can make, advantageously.

Fig. 132 is a diagram representing the exhausting and charging processes which take place at the end of each stroke of the piston, and it shows plainly the events occurring, from the time of the opening of the exhaust valve, until the compression of the charge by the working piston is in progress.

In order to promote the effective scavenging of the cylinder, the cylinder-head is formed as a deflector for the air current that enters through the admission valve on top of the cylinder. The object sought is to guide the general current of the charge in an axial direction so as to exclude, as much as possible, the intermingling of the new charge with the waste gases and to insure the latter's more effective expulsion.

Fig. 133 shows the general construction of the cylinder and

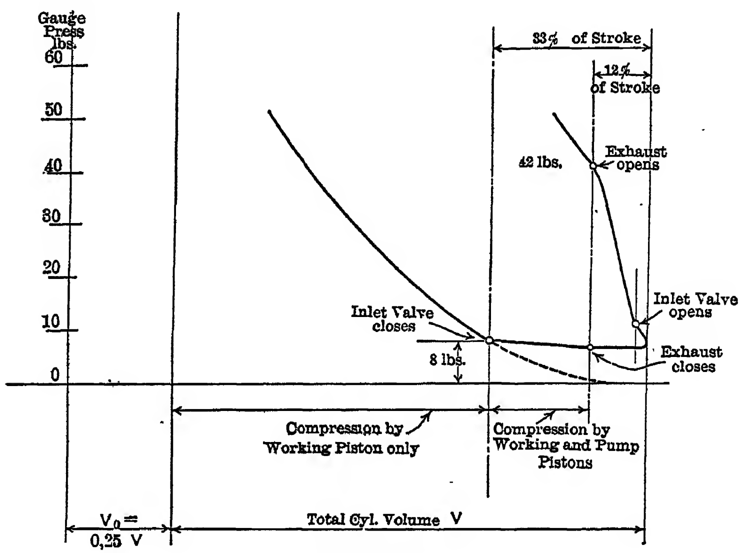


FIG. 132.

valve-gear of the Koerting engine built by the De La Vergne Machine Co. of New York.

The valve-gear shaft is driven, of course, in unison with the main shaft, by means of a pair of mitre-gears and the valves are actuated from this shaft by cams, and rockers which are fulcrumed on the valve-bonnets. The spiral gear for driving the governor will be seen in the illustration, at about the middle of the valve-gear shaft, but the governor itself is removed. The cooling system for the cylinder is plainly shown; the water being admitted at a low point at each end of the cylinder, and discharged from the very top of the middle of the same. It drains from there

through a pipe leading to a general cooling-water drain in the hollow base-casting, which discharges through the nozzle shown in the foreground in the illustration.

The piston is, of course, made hollow and is cooled by water admitted through a hollow piston-rod.

The engine is started by means of compressed air, which is admitted to the working cylinder by a small slide-valve, similarly as in an ordinary steam or compressed-air engine. The general

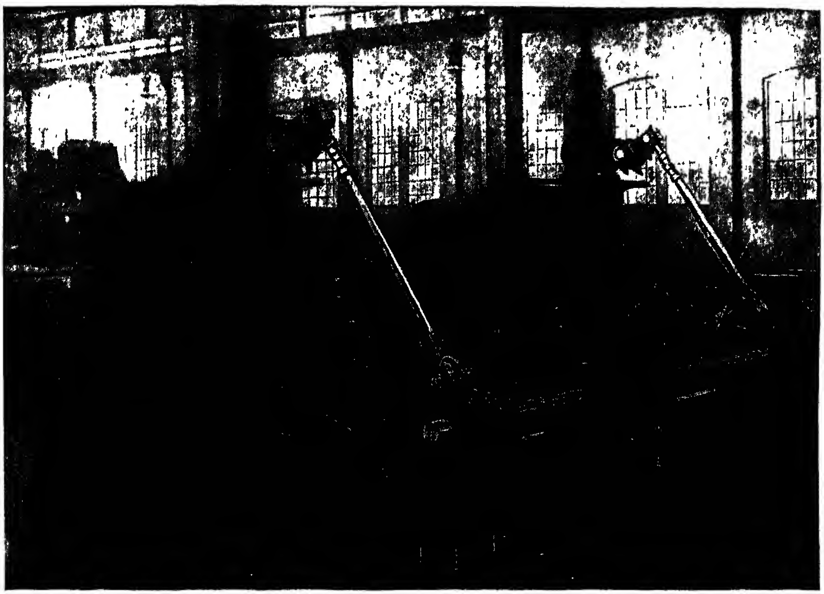


Fig. 133.

starting arrangement, which is practically the same for all types of double-acting engines, is described more in detail at page 404.

The ignition is controlled by means of an independent igniter shaft, the position of which, relatively to the valve-gear, can be advanced or retarded to suit the required time for firing.

The theory is often advanced, particularly with respect to the two-cycle engine, that a stratification of the different components of the charge in the cylinder will occur, so that after compression there will be found, nearest the piston, in succession, layers of

burned gases and of air, and from there the charge will successively grow richer and of a more perfect quality toward the inlet valve, near which the igniter generally is located. It seems, however, that the word "gradation" would more nearly express the occurrence; the likelihood being that the charge actually, under certain conditions, becomes more or less gradated to its composition—the inert elements and scavenging air being mixed in, in a greater quantity, toward the piston.

It is, of course, possible, by increasing the amount of air for scavenging, to increase the thoroughness with which the waste gases are expelled from the cylinder, though with increased loss due to negative work. It would, however, be with risk of wasting part of the fuel through the exhaust port that the attempt would be made to fill the cylinder completely with fuel-charge. On this account the cylinder is generally not charged with fuel-mixture at a higher rate than to about 85 per cent of the volume of its working stroke. The complete charge in the two-cycle engine will, therefore, be approximately at par with that of the four-cycle non-scavenging engine.

The Oechelhaeuser Engine.—Another two-cycle engine, which, like the Koerting engine, is used extensively in Germany, where it is frequently installed for the operation of blast-engines, particularly, is the Oechelhaeuser engine, illustrated in Figs. 134*a* and 134*b*.

A feature of this engine is that its cylinder is equipped with two pistons; one reciprocating in the front end of a long cylinder, while the other works, adversely to the front one, in the rear end. The rear piston is, by means of a rear crosshead and yoke, side-rods, and a double set of main crossheads and connecting-rods, connected to two side-cranks of the same throw as the centre-crank to which the front piston is connected. This feature is plainly shown in the plan view, Fig. 134*a*.

The engine-shaft is, accordingly, equipped with three cranks of which the outside ones take, each, one-half of the force due to the impulse on the rear piston, and the centre-crank takes the full force due to the impulse on the front piston. For each revolution of the wheel there occurs, thus, two simultaneous

impulses; the one on the front piston moving toward the front and the one on the rear piston moving toward the rear. This is in effect, as far as the power, and the speed regulation during the cycle are concerned, the same as if there were for each revolution only one impulse, acting on the given crank arm, of twice the magnitude as that due one piston. Assuming that the reciprocating parts of an engine of this type were of the same weight as those of a four-cycle single-cylinder double-acting engine, then the speed regulation during the cycle would for both engines be the same, and the engines would require the same weight of fly-wheel. The balancing of the reciprocating parts of the engines would, however, be materially different.

It will readily be seen that the balancing of the Oechelhaeuser engine is nearly perfect, due to the fact that the two pistons are always moving in an opposite direction to each other. The only discrepancy from perfect balancing arises because the acceleration of the reciprocating parts moving toward the front is not, at all times, exactly the same as that of those moving toward the rear, due to the influence of the limited length of the connecting-rods. There may, of course, besides be some difference in the actual weights of the two sets of reciprocating parts.

On account of the practically perfect balancing, any tendency toward the rocking of the engine on its foundation is eliminated, but the tendency toward noisiness of its many main pins is still there; particularly if the reciprocating weights should be heavier than suitable for the compression-pressure and piston-speed employed. The piston-speed used in this engine is generally from 800 to 900 feet per minute.

It will be of interest to examine somewhat closely into the construction of the engine illustrated, as such an examination will bring out several features of advantage belonging to this engine-type.

The cylinder is constructed, as may be seen in the sectional view, Fig. 134*b*, of two plain inside cylinder-liners over which the two jacket-castings have been forced and connected by heavy flanges, at the middle of the completed cylinder. The inside flanged ends of the two liners butt together, and are held securely

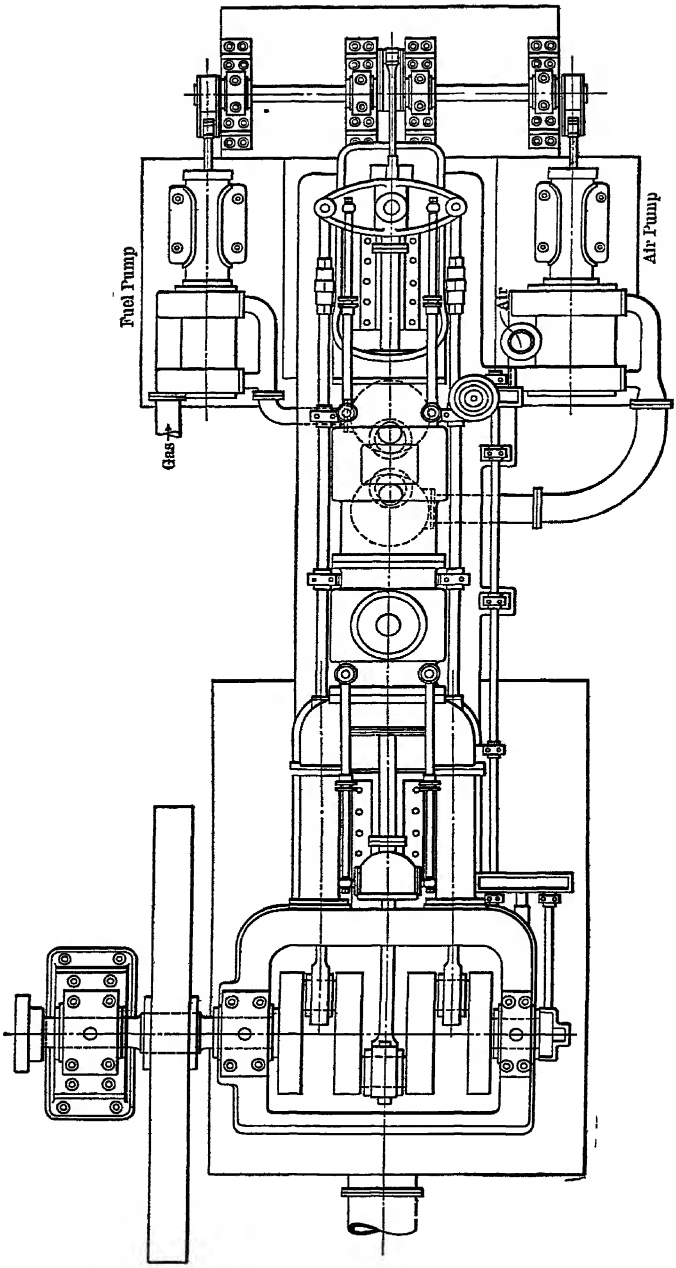


FIG. 134*a*.—Oechelhaeuser Engine. Plan View.

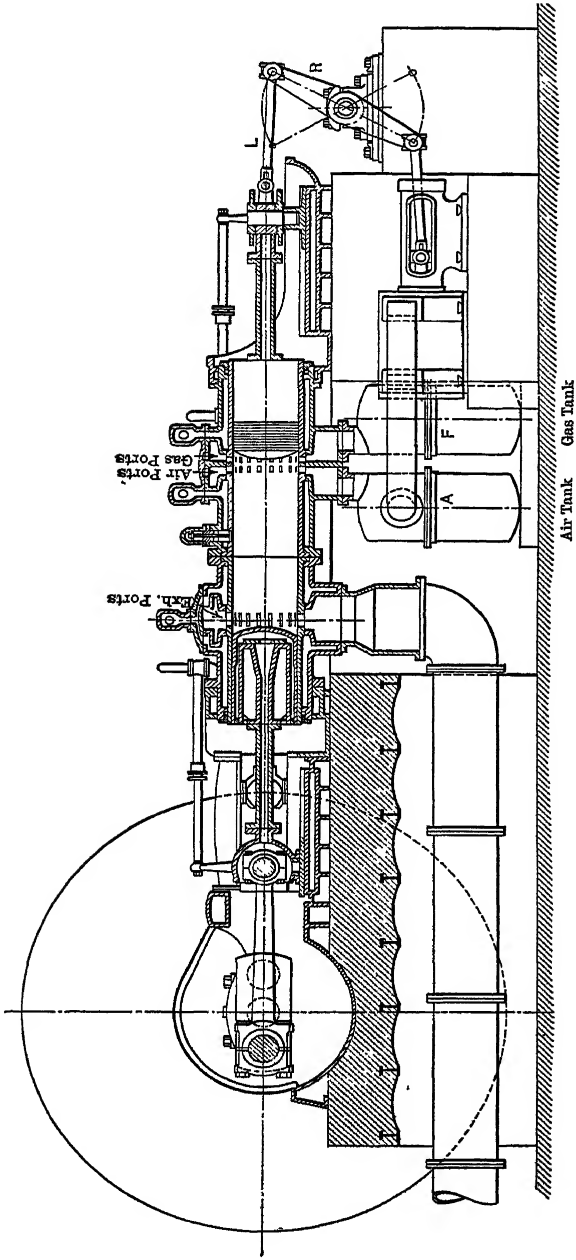


FIG. 134*b*.—Oechelhaeuser Engine. Sectional Elevation.

by the same joint that connects the jacket-castings, while their outside ends are free to expand and contract independently of the jacket, and, to allow this freedom of the liners, the water-joints at the outside ends of the cylinder are made by means of a packing, and by a gland which draws up against the ends of the outside castings.

The rear half of the jacket has, cast integral with it, the air and gas admission-channels, while the front half contains the discharge-channel for the waste gases. All these channels encircle and cut through the water-space of the jacket, so as to communicate all around the cylinder with rectangular ports which are cut in the liners, near the ends of the actual working space of the cylinder. At the bottom of the cylinder the channels connect, respectively, with the air supply, with the gas supply, and with the exhaust pipe.

At the middle of the cylinder there is, as seen, applied a check-valve, through which the compressed air for the starting of the engine is supplied.

The cylinder, thus completed, is centered at the front end, in the main engine frame and, at the rear end, in the rear guide-frame, and held axially by flanged joints. It will be evident that, as there will be no axial strain in the cylinder due to the working pressure on the pistons, there will never be any question as to the strength of any transverse section or circumferential joint between parts of the construction; and, as no strains are transmitted through the framework of the engine, there will be no difficulty of holding the rear part of the engine solidly to its foundation. These advantages are, most particularly, some of those belonging to this type of engines.

The pistons are water-cooled; the water-space being connected through hollow piston-rods, by means of telescoping tubes to the supply and discharge water-system.

In order to facilitate the removal of the pistons from the cylinder, there are inserted, between the crossheads and the pistons, flanged, short piston-rods that can readily be disconnected, and removed, in order to accommodate with the required space for sliding out the pistons.

The rear crossbar which connects the two side-rods is attached to the piston-rod by means of a pin-joint, which allows it to swivel the amount necessary to equalize the strains in the side-rods, and, thus, also the pressure on the side-cranks.

As in the Koerting engine, the cylinder of the Oechelhaeuser engine is scavenged with air, which in this engine is supplied from an air-tank, *A*, located directly underneath the cylinder. The fuel supply is also furnished from a similar tank, *F*. The air and the fuel supplies to these tanks are kept up by means of the air-

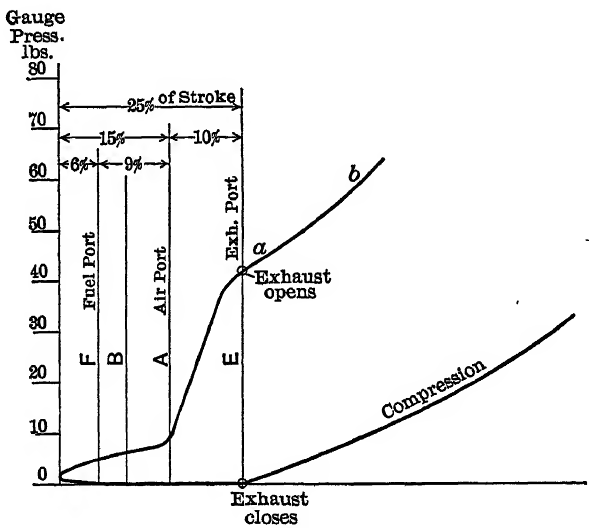


FIG. 134*c*.

and gas-pumps, which are both, generally, driven from the rear crosshead. In the engine illustrated they are driven by means of a rocking shaft connecting to the rear crosshead by means of the rocker, *R*, and the link, *L*.

It will be evident how the pistons, by covering and uncovering the inlet and exhaust ports, will act as valves for the admission and release of the charge. The timing of the various events of exhaust, scavenging, fuel admission, and so forth, will most readily appear from the diagram, Fig. 134*c*. The lines *E*, *A* and *F* represent the cutting edges of the port-openings. The cutting

edge of the exhaust port, *E*, is approximately 25 per cent, the cutting edge of the air port, *A*, 15 per cent, and that of the fuel port, *F*, 6 per cent of the piston-travel from the end of the stroke. The ordinates of the diagram represent the approximate pressures existing at the various points of the crank-end of the pressure-card. The line *a–b* is the lower end of the expansion line. When the piston passes the line *E* the exhaust port becomes uncovered for the release of the old charge, which, after a period corresponding to about 10 per cent of the stroke, becomes of a pressure equal to, or below, that carried in the air-tank. At *A*, therefore, approximately 15 per cent from the end of the stroke, the ports communicating with the air-tank will uncover and the admission of the scavenging air commence. The admission of pure air for the displacing of the old charge is continued during a period of about 9 per cent of the stroke, until the fuel-port is uncovered, which occurs when the piston has arrived within 6 per cent to the end of its stroke. During the remaining part of the stroke, and until the fuel-port becomes covered during the return-stroke of the piston, air and gas, in a suitable proportion for a combustible mixture, are admitted. After the admission of the fuel-charge is completed, pure air only will again be admitted during a period corresponding to a piston travel of 6 per cent of the stroke, from *B* to *A*, until the air ports become covered. The exhaust has remained open from the time the piston, going out, at *E*, uncovered the ports, until they again become covered at the return of the piston to *E*. From there compression of the charge will continue to the end of the compression-stroke.

The main idea of this charging process, it will appear, is to obtain an effective mixture at the middle of the cylinder, where the ignition of the charge takes place, and to enclose this mixture at the ends of the cylinder, toward the pistons, by bodies of air.

It is evident, that should an overload be thrown on to the engine causing it to slow down, then, if the charging pressure were too high, the charge may have time to reach the exhaust ports, before these become closed by the front piston, and not only become wasted, but liable to cause premature ignition when ignited by the heat of the exhaust pipe. To preclude pre-ignitions

there is often applied in the exhaust-pipe, close to the cylinder, a jet of water that helps to keep the ports cool. In order to cool and effectively displace the hot neutrals, the air for scavenging is often carried of a quite high pressure (from 6, sometimes to 9 pounds). The higher the pressure the more effectively the air, at its expansion to the discharge pressure, will chill the waste products.

The relative proportions between the air and the gas admitted for the actual charge, at normal load, must, of course, be carefully adjusted to that suitable for the best mixture. To effect such a proportioning, even though the air in the tank, for scavenging purposes, is carried at a high pressure, there is often applied in the outlet from the tank a throttling valve, which, by means of an eccentric on the valve-gear shaft, is operated so as to throttle the air-pressure to that suitable for the mixture, as soon as the gas ports are ready to open. This arrangement is used by Borsig, who is a well-known builder of this type of engines.

The regulation of the Oechelhaeuser engine is arranged differently by different builders. The general idea is to effect, as nearly as possible, a constant-quantity mixture for all load-conditions. To accomplish this it would be required that the fuel-charge, only, be throttled, and by-passed to the suction side of the fuel-pump, more or less, to suit the load, but under such conditions the charge may, at light loads, become too lean even to ignite, and for that reason the air also must, generally, be throttled and by-passed by the governor.

The Borsig-Oechelhaeuser engine is, as a rule, fitted with two ring-valves surrounding the cylinder-liner, at the air- and at the gas-ports, and having port-openings registering with those in the cylinder. By means of these valves the air- and gas-ports may be adjusted, both by hand, to suit the full-load conditions, and by the governor, for the throttling of the charge at light loads. The air valve is arranged so as to throttle the air-ports at the top of the cylinder, principally, by which arrangement the charge becomes richer near the igniters, and the firing more sure.

An engine of this type of a cylinder diameter 43.3 inches and 53.2 inches stroke, running 100 revolutions per minute on coke-

oven gas, is rated by the builders at 1,500 B.H.P., which is a very conservative rating. For an overload capacity of 20 per cent the engine would be of, approximately, 2,100 maximum I.H.P. The corresponding suction-displacement of the piston, per horse-power, is thus as much as 4.4 cubic feet per minute, or the mean effective pressure 54 pounds per square inch of the piston. The test of the Borsig-Oechelhaeuser engine included in Table XXXI records, however, an average mean effective pressure of 74 pounds. Good coke-oven gas, being of a heating-value practically the same as that of illuminating gas, should readily give a mean pressure even higher than this. However, as its heating-value generally fluctuates materially, a conservative figure should be used at estimates. The figure 74 has been recommended in Table XI for coke-oven gas.

The work required by the air and gas pumps is generally from 10 to 14 per cent of the total work indicated by the working cylinder, and this work, together with other resistances of the engine, must, of course, be deducted from the total indicated work obtained in the working cylinder for obtaining the brake horse-power of the engine.

The Indicated Power of the Two-Cycle Engine.—The question has been raised, in case a compressor-piston for supplying the compressed charge is driven by a two-cycle engine: should the total indicated power of the gas-engine cylinder be considered the indicated power of the engine; or should the power required by the compressor be deducted from the total power shown by the indicator-diagram, in order to obtain the indicated power of the engine?*

In the latter case, the indicated power would be at par with hat obtained in a four-cycle engine-cylinder, which effects entirely its own compression. The matter resolves itself into a question as to whether the indicated power of the engine is that indicated by the gas-cylinder alone, or whether it is that indicated by the gas-cylinder and the compressor cylinder together; the latter being negative.

* For the various opinions expressed on this question see *Zeitschrift des Vereins Deutscher Ingenieure*, for February, April, and May, 1905.

The value of the indicated power, or of the efficiency of an engine, is, of course, ascertained in order to serve as a basis for comparison with results obtained from other engines. In order that the results obtained from this special type of two-cycle engine shall be comparable with the condensing steam-engine, with the Diesel engine, or with some types of obsolete gas-engines, the indicated power must be that obtained by the gas-engine cylinder, without reduction for the work of compression, but in order that they shall be comparable with results from the four-cycle gas-engine the indicated power must be that indicated by the engine as a whole.

To compromise these requirements, it has been suggested by Mr. Diesel to express the power obtained in a two-cycle engine according to the following definitions:

The *total* indicated power (of the cylinder) = the full power represented by the indicator-diagram of the working cylinder.

The *net* indicated power (of the whole engine) = the indicated power of the working cylinder less the pump cards.

The brake horse-power = the work delivered to the end of the engine-shaft, and which can be taken off by means of a brake and brake-wheel.

The efficiencies would be:

The total mechanical efficiency = the brake horse-power divided by the total indicated power (of the cylinder).

The dynamic efficiency = the net indicated power (of the whole engine) divided by the total indicated power (of the cylinder).

The compressor-factor = the power represented by the pump cards divided by the total indicated power.

The engine-friction = 1 − the dynamic efficiency.

Four-Cycle Engines.—The Otto Engine.—The Otto engine is an old and well-known four-cycle engine operating on the hit-or-miss principle. The governor is of the fly-ball type, and it controls the admission or exclusion of the fuel by shifting the inlet cam-roller on or off the cam, accordingly as the speed of the engine is below or above the normal.

The inlet- and exhaust-valves are arranged, one on each side of the combustion-chamber, in separate removable valve-casings,

and they are both operated by cams from a cam-shaft running along the side of the engine. This shaft is driven by means of a pair of spiral gears from the main shaft, at one-half the speed of the latter, and it carries besides the two valve-cams also, on

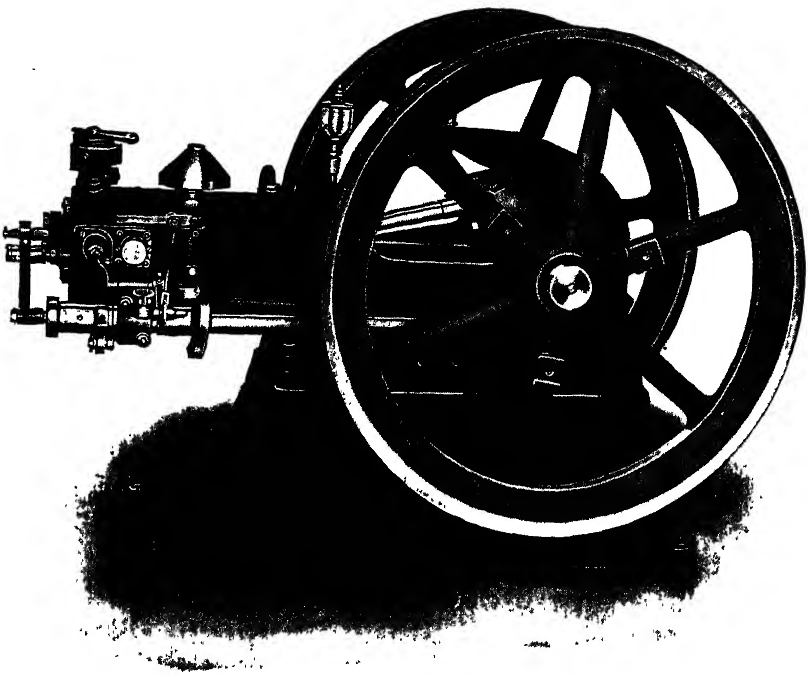


FIG. 135.

its extreme rear end, a small crank for operating the make-and-break ignition.

A fair idea of the general construction of the engine can be obtained from the illustration, Fig. 135.

The governor is shown in detail in Fig. 136, and the inlet valve-rocker carrying the cam-roller will be seen at the lower part of the figure. The position of the cam-roller is controlled by the governor by means of the bell-crank, in such a manner, that when the governor-sleeve is down then the bell-crank brings the cam-roller in a position to meet and engage with the inlet cam,

and thus effect the lifting of the valve. When, on the contrary, the sleeve, at excessive speed of the engine, rises to a certain point, then the cam-roller will be brought by the bell-crank out of the position for meeting the cam, which, consequently, will cause the inlet valve to remain closed during one or more strokes of the piston.

The Modern Four-Cycle Throttling Engine.—Figures 137, 138, and 139 are the Plan View, the Front Elevation and the Longitudinal Section of a common type of four-cycle throttling engine, which has been used with particular success for producer-gas. The views represent an engine originally designed by Mr. Max Munzel, late of the firm G. Luther of Braunschweig, Germany, and they afford good illustration for the study of the details of engines of this type.

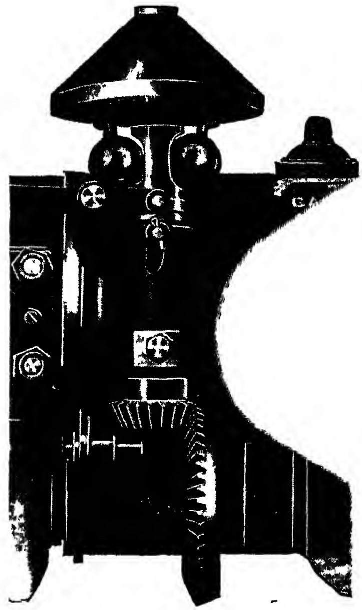


FIG. 136.

Fig. 139 shows the main frame and the cylinder jacket to be cast in one piece, and the cylinder bushing to be inserted in the jacket-casting and held in place by a flange at the rear end; thus leaving its front end free to expand or contract without throwing strains on to the main casting. The inlet- and exhaust-valves and the igniter are located in a cylinder-head casting that contains the main part of the combustion-chamber, and which is thoroughly water-cooled all around. The exhaust-valve is readily removable by removing the inlet valve-bonnet, and it is seated on a removable steel bushing.

The cam-shaft is driven at one-half the speed of the main shaft by means of a pair of spiral gears located in an oil-tight gear-casing, as shown in Figs. 137 and 138, and the governor is also driven by the same means, similarly located. The inlet

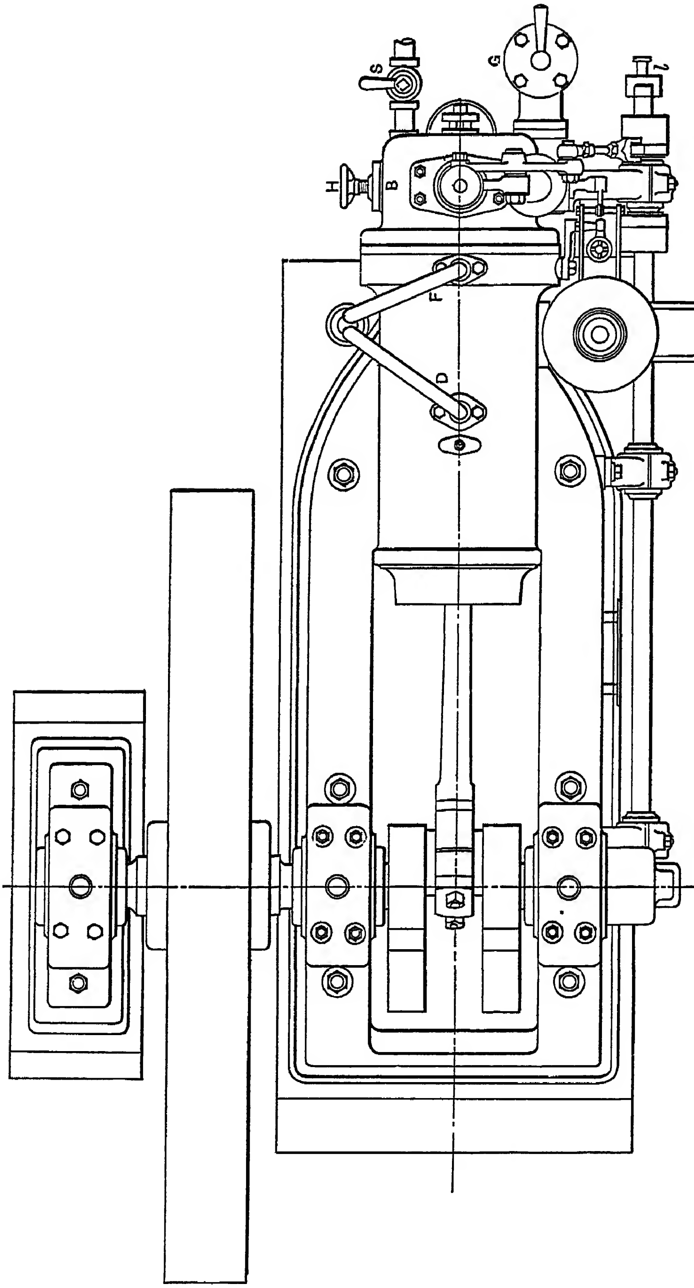


FIG. 137.—Four-Cycle Single-Acting Gas-Engine. Plan View.

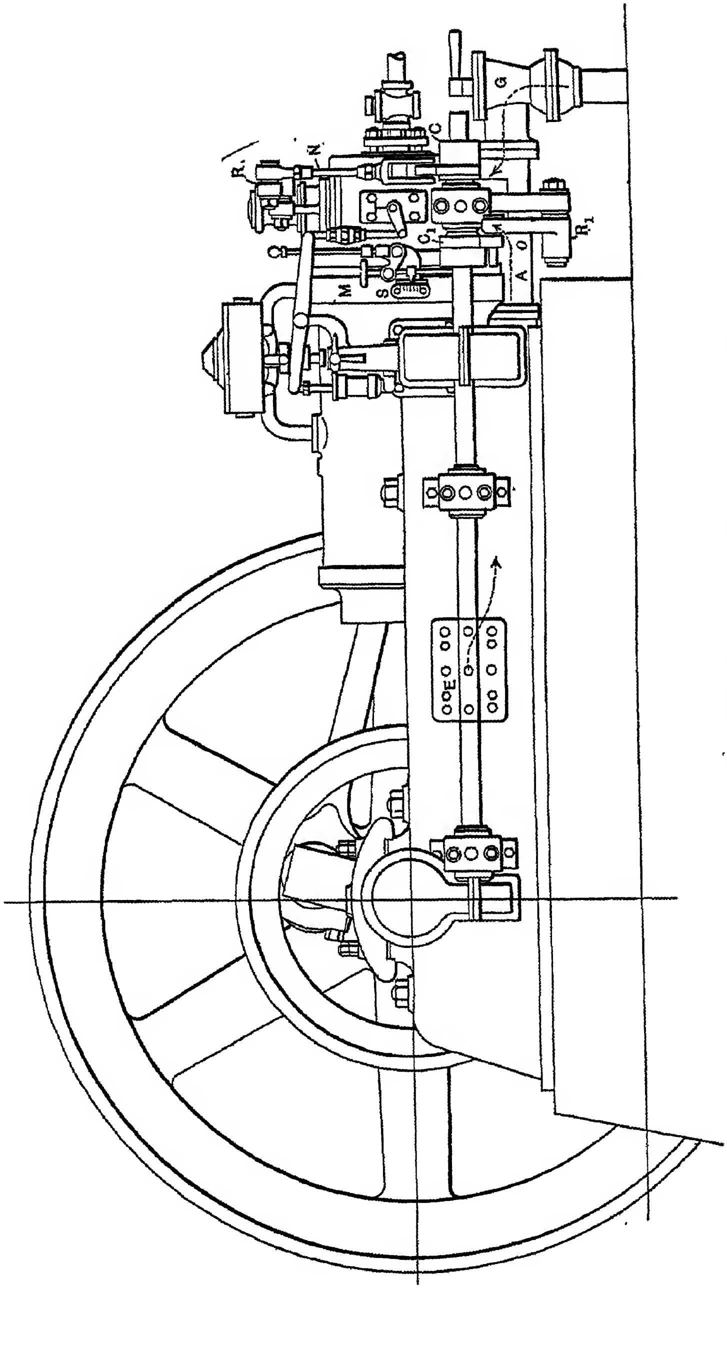


FIG. 138.—Four-Cycle Single-Acting Gas-Engine. Front Elevation.

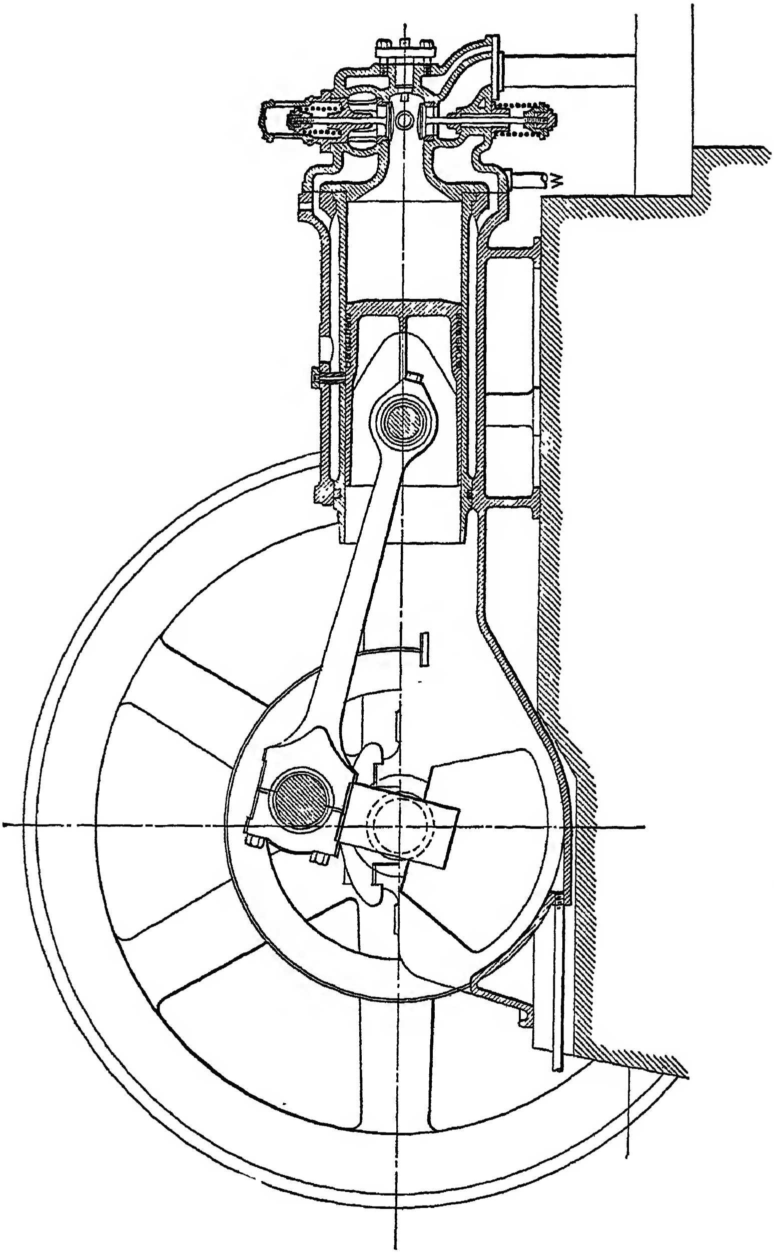


FIG. 139.—Four-Cycle Single-Acting Gas-Engine. Section.

valve is actuated by means of the cam C, valve-rocker connection N, and valve-rocker R. To relieve the compression, at the time of starting the engine, the exhaust cam-roller o, can be slid sideways off the regular exhaust-cam, C_1, and on to a smaller starting-cam located at the side of the cam C_1, which gives less than one-half the regular compression-pressure.

The engine draws its charge of air through the frame, which helps to keep the engine cool. The air enters through some holes back of the plate E and is drawn in to the mixing-chamber through the pipe A, while the gas is drawn through the gas valve G; and the gases are mixed immediately below the throttle valve T. The proportioning of the mixture is done by opening or closing a throttle-valve located in the air-pipe A, by means of the small hand-wheel M; and the position of this throttle is indicated by an index running over a scale s. By running this index slightly up or down the scale when the engine is operating, and noting the effect the various positions of the throttle will have on the speed, or on the sound of the engine, the best proportioning can readily be ascertained.

The ignition of the charge is effected in this engine by means of the magneto ignition illustrated and described at page 332; the pick-blade lever L attaching to the crank l, Fig. 137.

The cooling water is admitted to the jacket at a lower point, at W, and it is drained from two points D and F at the top of the cylinder; each drain having a separate regulating-valve. The object with this arrangement is to allow the regulation of the temperature of the water in the combustion-chamber jacket independently of that in the cylinder-jacket; still having both jackets connected.

The starting of an engine of this type is generally accomplished by means of compressed air, which is supplied through the starting air valve S, and admitted to the cylinder through a check-valve located back of the bonnet B. The hand-wheel H is for the purpose of locking the check-valve to its seat during regular running. The position of this valve, relatively to the main valves of the engine, may be seen to better advantage in Fig. 105, page 307, which represents a cross-section through the combustion-

chamber of an engine practically the same as the one being described, excepting that the governor throttle valve is, in Fig. 105, put as close to the inlet-valve as possible. The advantage of this latter arrangement is that the mixing of the air and the gas is effected closer to the inlet-valve, and, hence, the volume of the mixing-chamber is cut down. This feature may be quite desirable when the fuel is of the kind that is apt to cause back-firing into the mixing-chamber.

Manipulation at the Starting of the Four-Cycle Engine.—To prepare for the starting of the engine the main crank must

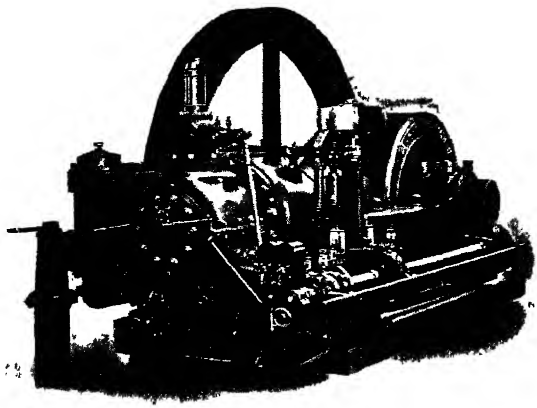


FIG. 140.—Koerting Four-Cycle Engine.

be put in its position for a firing stroke, and a little above the head-end centre; the inlet- and exhaust-valves then being both closed. The hand-wheel *M* is adjusted for a proper mixture, the exhaust cam-roller put on the starting cam, and the hand-wheel *H* turned so as to relieve the air check-valve. The leads from the magneto are then connected to the spark-plug, and the ignition retarded so as to fire late.

To start the engine the gas is turned on and the air-starting valve *S* is opened to admit air-pressure to the cylinder, and kept open until the crank has started to move, then quickly closed. During the next following forward stroke, the engine will draw its regular charge, which will be fired at the end of the return

compression-stroke. The engine-piston, thus, is called upon to make one exhaust-stroke, one suction-stroke and one compression-stroke due to the impulse from the air-charge. The air-pressure, therefore, is required to be quite high—100 to 150 pounds is variously used, depending on the compression pressure employed.

An alternate method for starting would, of course, be to give the engine several impulses by compressed air, to bring it up to speed before the gas-valve is opened. When the start is made the exhaust cam-roller is put over on the regular cam for normal compression, the ignition is advanced to bring the engine up to normal speed, the lubricators are attended to, and the cooling water for the jackets turned on.

The Koerting Four-Cycle Engine.—An engine of very much the same general construction as the Munzel engine is illustrated in Fig. 140, and the arrangement of its valves is shown in the transverse and longitudinal sections through its combustion-chamber and valve-casings, Figs. 141 and 142.

When engines of this type are connected together to one shaft as twin engines, with a common fuel supply, or when two engines draw from a common gas-main, it will be of advantage to apply in the fuel-supply port of the engine a check-valve that will prevent one engine from drawing the charge back from the other. Such a valve is applied in the Koerting engine and it serves also as a mixing-valve for separating the air and the fuel, until the charge is drawn in by the engine.

The gas and the air arrive at the mixing-valve through the pipes marked, respectively, *G* and *A*, in Fig. 141. When the valve, which covers both the gas and the air ports, is raised, due to the suction of the engine, the gases will flow as indicated by the arrows, and mix in the chamber immediately above the valve. After the suction-stroke is completed, the valve will seat itself and prevent any back flow of the fuel-mixture. In seating, the valve is cushioned by the dashpot formed in the bonnet of the mixing-valve chamber. This dashpot becomes accessible for adjustment by removing the protecting hood placed over it. The governor throttling-valve is, as will be seen, located between the mixing-valve and the main inlet valve.

The Olds Engine.—The Olds Gas Power Co. of Lansing, Mich., build a double-throw-crank engine of the same general type as the ones just described; and, as far as its main parts are concerned, also of very much the same construction. Its valve mechanism is, however, materially different from those of the Munzel or Koerting engines, and will warrant a special study.

The valve gear is shown, fully, in detail in the illustrations, Fig. 143, which is a section through the combustion-chamber and valve-casings, and Fig. 144, which is a side elevation of the head end of the cylinder, looking from the valve-gear side. Referring to Fig. 143, it will be seen that the motion for the admission valve is derived from the cam-roller *R*, through the valve-lever connection *C* and valve-lever *L*. As in the Deutz valve gear described previously, this lever is not hinged to a fixed fulcrum, but the fulcrum, against which the lever is forced in opening the valve, consists of a movable block *B*, the position of which is controlled by the governor. The governor-lever *G*, it will be seen, controls the motion of the arm *A* to which is linked the block *B*. Hence, at a slow speed of the engine the fulcrum-block will be moved outward, away from the valve-end of the lever, causing a high lift to be given to the admission valve, and at a high speed the block will be moved inward, causing the lift of the valve to be reduced.

The admission valve consists of the main inlet valve and of the gas valve. The latter being a double-ported piston-valve sleeve, which is secured concentrically on the inlet valve-stem. The inlet valve and the gas valve will thus move together as one, but the gas valve, having a certain amount of lap, will not open the gas port before the main valve has been open some little time for the admission of air, only, from the air-passage, and it will close the gas port before the main valve closes against the admittance of air. The object with this arrangement is to scavenge the valve-casing from any explosive mixture, which precaution makes the engine safer against back-firing into the valve-chamber. Furthermore, the regulation becomes somewhat on the order of a constant-quantity regulation, which appears desirable for certain fuels.

It will readily be seen, if we assume that the admission valve, at a very light load, is lifted hardly enough to uncover the lap of the gas valve, that, then, air only would be admitted during the suction stroke. By opening the valves a little further principally air, and only a little gas, will be admitted; and so forth. And at the full lift of the valves a normal charge will result. Showing that, for decreasing loads and decreasing valve-lifts, the charge will grow more diluted. Of course, the charge will also become of less density.

The gas stop-valve *V*, it will be seen, consists of a cylindrical sleeve having a port cut through it at one side, by which, by revolving it, the gas may be throttled more or less, or cut off entirely. A more permanent proportioning of the air in the mixture may be made by a butterfly valve which is applied in the air passage.

The valve gear illustrated, which is for a cylinder somewhat more than 21 inches in diameter, is equipped with a water-cooled exhaust valve; the cooling water being supplied at *a*, and drained through the small central tube, inside the valve-stem, which terminates at *b*.

The illustrations, Fig. 143 and Fig. 144, show the engine to be fitted with an automatic air-starting device. This feature is shown at *D*, Fig. 144, and in detail in Fig. 145.

Instead of operating the air-starting valve by hand, as described at page 363, the valve can, of course, be automatically opened and closed. This is done in the present engine by means of the small bell-crank and cam-roller shown at *E*, Fig. 143; the cam-roller being actuated by a special starting cam *e* on the end of the cam-shaft. After the engine has been put in a position for starting, which is with the crank a few degrees above the head-end centre and with the inlet- and exhaust-valves closed, the small cam *S* on the air-starting handle, Fig. 145, is forced against the valve-stem, which has for effect to open the valve. At this time the cam *e* is underneath the cam-roller *r*, but at the proper time for the closing of the air valve the roller will ride off the cam and allow the valve to close. The air valve may thus be worked continuously by the air-starting cam *e*, for several turns of the

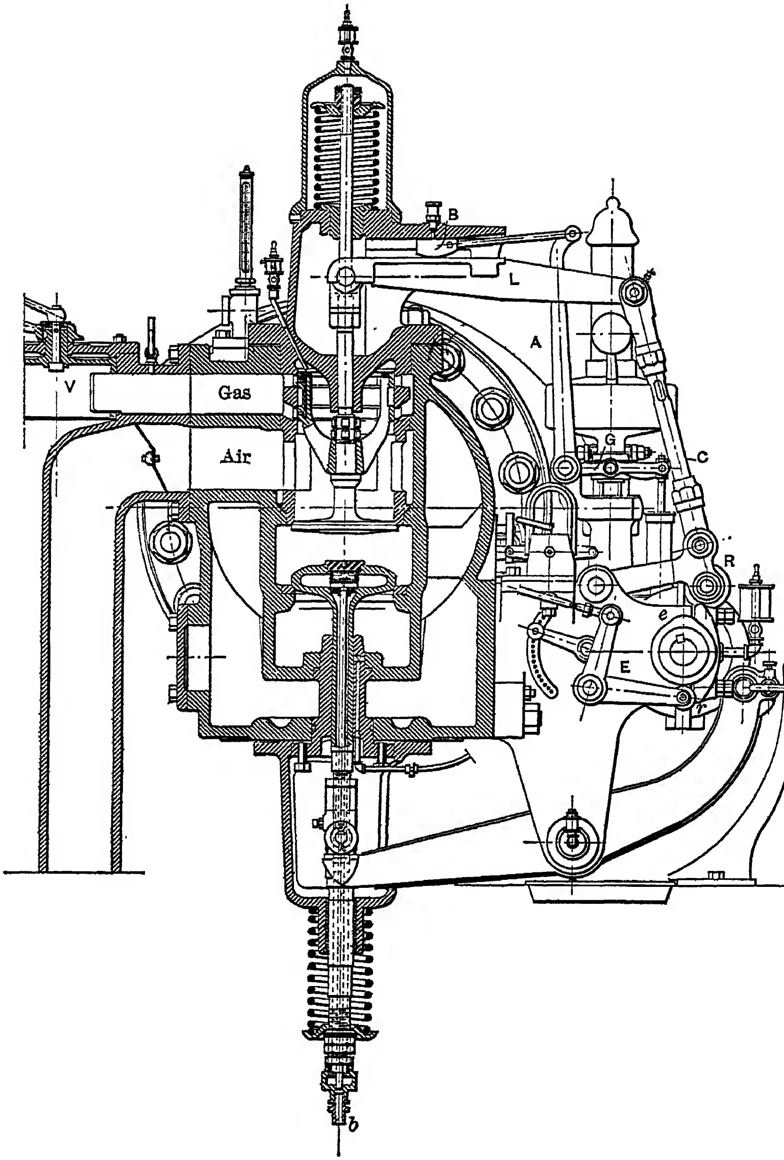


Fig. 143.—Olds Valve Gear. Section through Combustion-Chamber.

FIG. 145.

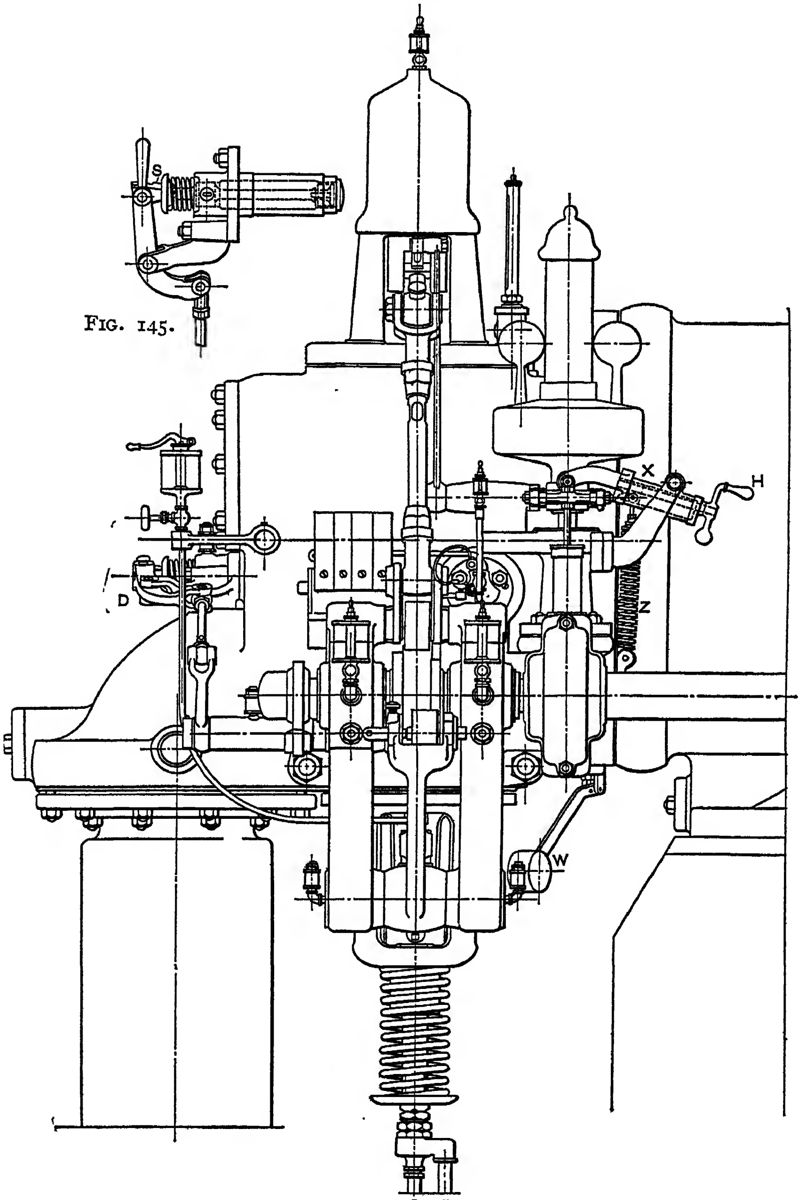


FIG. 144.—Olds Valve Gear. Front Elevation.

engine, until the cam *S* on the starting handle is withdrawn from contact with the end of the valve-stem, by the shifting of the handle to a position in a right angle to that it occupies in the figure. With the cam *S* withdrawn, the starting-valve will be closed and the roller *r* hang fire of the starting cam.

The combustion-chamber is, at the bottom, provided with a small drain and relief valve, which is counterweighted by the weight *W*.

By means of the handle *H*, the speed of the engine may be varied within some few turns. The leverage with which the spring *z* acts on the governor-sleeve can, namely, be changed, within the limits of the slide *X*.

Multiple-Cylinder Engines.—Open-end cylinders below 21 inches are not generally equipped with water-cooled pistons, as the atmosphere, due to the motion of the piston, will have adequate access for cooling it. Neither is, as a rule, the exhaust-valve of cylinders below that size water-cooled. An engine of a single 21-inch cylinder, running at normal speed, will on producer-gas develop, on an average, 125 B.H.P.; or if constructed as a two-cylinder engine twice this power will be obtained.

For installations below 250 to 300 B.H.P., therefore, the single-acting four-cycle engine becomes, on account of its simplicity, a very serviceable type. Two-cylinder engines are by some builders arranged as twin engines; others prefer the tandem arrangement, but as far as the regulation or the required weight of wheel is concerned one type is as favorable as the other. The space available for an installation may, however, in a great measure become a determining factor as to which type will be the most suitable in a special case.

As to twin engines; any of the four-cycle engine types described in the preceding may, when occasion requires, be arranged as such, simply by coupling together two engines on one shaft having both cranks in line.

The Jacobson Tandem Engine.—Fig. 146 illustrates a type of tandem engine built by the Jacobson Machine Co., of Warren, Pa. The view, which is looking from the valve-gear side of the engine, shows the valve-gear lay-shaft to be driven by a pair of spiral

gears from the main engine-shaft, and its speed is, of course, one-half the speed of the latter. The exhaust-valves are, as the figure plainly shows, operated by cams, while the inlet-valves are actuated by means of eccentrics.

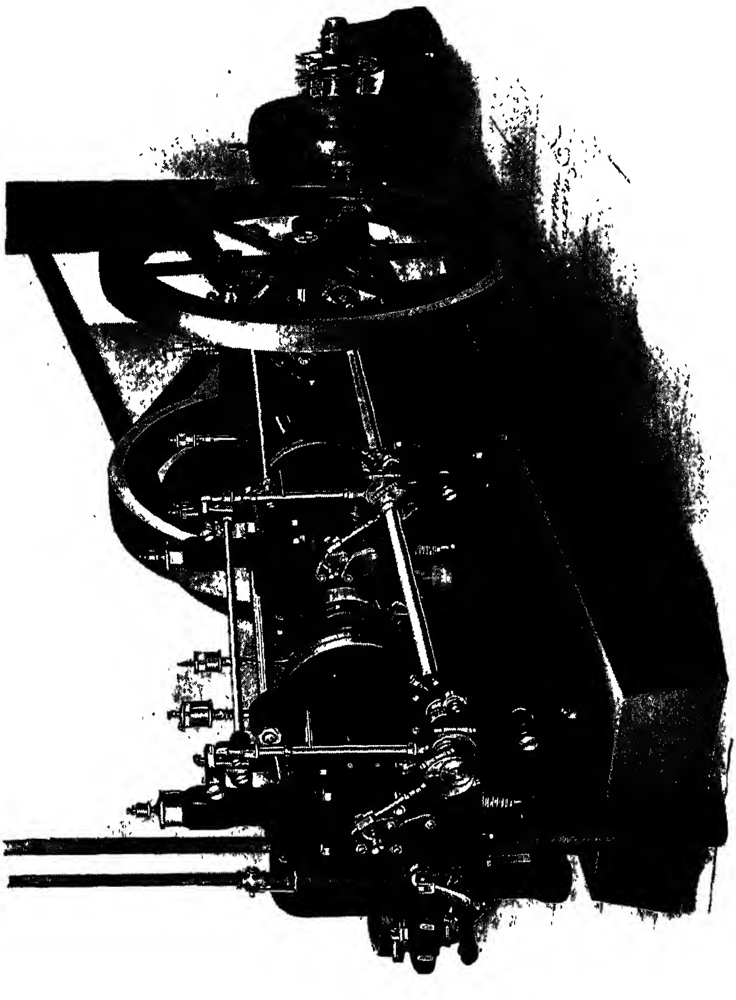


FIG. 146.

The valve gear is shown in detail in the sectional view, Fig. 147, and it will be observed that the regulation is effected by means of a release gear by which the inlet-valve is disengaged and closed at varying points of the stroke, to suit the load. To

some extent the cylinders are scavenged with pure air in order to prevent, as much as possible, any back-firing due to slow-burning mixtures.

The operation of the gear is as follows:

The inlet valve *I* is opened by means of the rocker *R* when the pick-blade *P* on the end of the eccentric rod *E*, lifts the end of the rocker-arm, and the valve becomes opened more and more, until the roller *D* rides up on the trip-block *T*, the position of which is controlled by the governor. It is evident that the further the trip-block is moved to the right the earlier in the stroke the valve will trip.

As soon as the main inlet valve opens, air will be admitted to the cylinder from the air port, for scavenging purposes. The gas valve *G* has a sliding fit on the main valve-stem and is held closed by the spring *Z*, until the collar *C*, which is solid on the valve-spindle, moving down, forces the valve to open against the spring-pressure back of it. The position in which the valves are drawn is such, it will be observed, that the main valve is opened to some extent for the admission of air, and the collar *C* is just touching the hub of the gas valve to effect its opening, in unison with the main valve, as the latter is further opened. *S* is the main valve-spring, the function of which is to close the main inlet valve, and to hold it closed against the partial vacuum that will be created in the cylinder at light loads.

For the cushioning of the valves, in seating, a cushioning-pot and piston are provided at *O*. Means are also provided in the gas and air ports, whereby the proportioning of the air and gas may be regulated to suit any kind of fuel that may be used.

The Premier Engine.—A well-known English tandem engine is, in sizes above 150 horse-power, built by the Premier Gas Engine Co., at Sandiacre, Nottingham, Eng. The arrangement of its cylinders and general construction is the same as that of the engine described above, excepting that the Premier engine is positively scavenged. The main crosshead, working in a bored guide-cylinder, serves the double purpose of guiding the end of the piston-rod and, as a piston, for compressing the scavenging air.

A test of an engine of this make running on Mond gas is reported by Mr. Humphrey in the Proceedings of the Institution of Mechanical Engineers, vol. 1901, page 79, and the principal figures of this test may be found in Table XXXI. The main sizes of the engine are: the working cylinders 28⅛ inch diameter

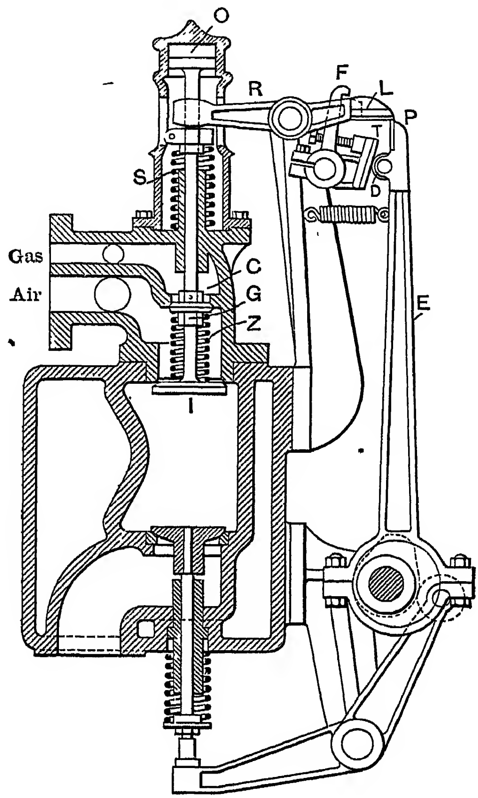


FIG. 147.—Jacobson Valve Gear.

x 30 inch stroke, the pump cylinder 43½ inch diameter x 30 inch stroke. The pistons are water-cooled, and to this fact, as well as to the positive scavenging of the cylinders, may be ascribed the very high mean effective pressure obtained. The test referred to reports an average mean effective pressure of 107 pounds. Figured from the power developed and the displacement volume

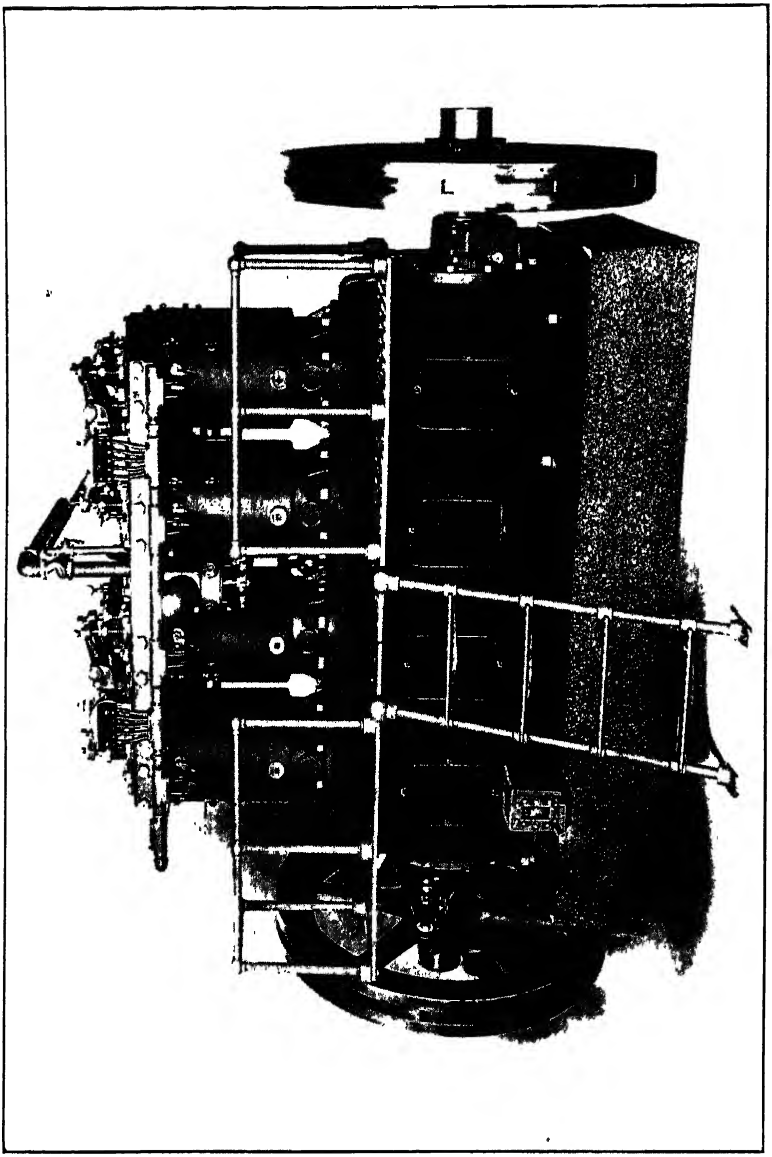


FIG. 148.—Bruce-Macbeth Four-Cylinder Engine.

of the pistons, the average mean effective pressure will be 81 pounds; to which, however, there should be added the pressure corresponding to the negative work of the pump piston to get the actual effective pressure in the working cylinders.

An average mean effective pressure as high as 81 pounds is not generally obtained with 144 B.T.U. gas in the non-scavenging engine, excepting for short periods of particularly suitable and steady load.

Vertical Multiple-Cylinder Engines.—In Figs. 148 and 148*a* is illustrated a type of multiple-cylinder gas-engine which is extensively installed, in units from 30 to 400 horse-power, for lighting as well as for motive-power purposes. This engine-type is built of two, three, or four cylinders, hence, with only a few cylinder sizes, a great variance of power can be obtained; and, it being of a very compact arrangement, it is particularly suitable when the space for an installation is limited.

From the illustration, Fig. 148, which is a reproduction of an engine built by the Bruce-Macbeth Engine Co., of Cleveland, O., in units of 100 horse-power up, a fair idea may be gained of the general construction of an engine of this type. The gas-supply elbow, the governor throttling-valve, and the mixture-supply chamber communicating to the various cylinders are plainly seen in front of the cylinders, whereas, correspondingly to the supply chamber, in front, there is, back of the cylinders, an exhaust manifold, which leads directly from the exhaust-valves, through a muffler, to the atmosphere. The inlet- and exhaust-valves are actuated by means of four cam-rockers, and cams on the cam-shafts located between each pair of cylinders. The transmission for motion from the main engine-shaft to the cam-shafts is plainly seen in Fig. 148*a*, which is a section through one cylinder and the crank-casing. The same figure shows also the governor, which controls the throttling valve by means of a long lever fulcrumed between the two middle cylinders, as well as the exhaust manifold referred to above.

At the left side of the crank-chamber, Fig. 148, there is shown a small generator driven by means of a belt from the fly-wheel hub, and it supplies the current for the high-tension spark-plugs

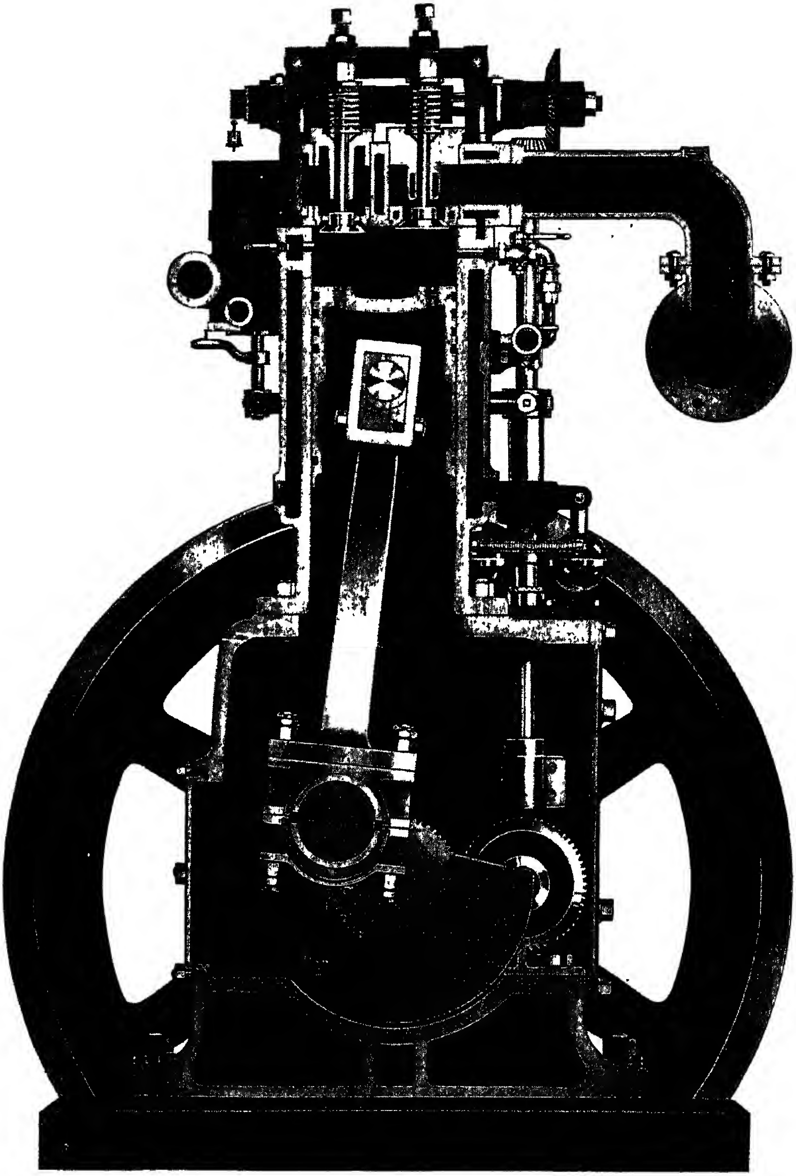


FIG. 148*a*.—Bruce-Macbeth Four-Cylinder Engine. Section.

fitted, in duplicate, to each cylinder. One set of spark-plugs may be furnished with current from another source than that of the generator shown, so as to make the proper ignition doubly insured.

One feature of the four-cylinder four-cycle engine, which recommends itself very much for large engines, is that its starting becomes very convenient. One air starting-valve is furnished for each combustion-chamber, the operation of which is effected by cams on the main cam-shaft, and so timed that the air ad-

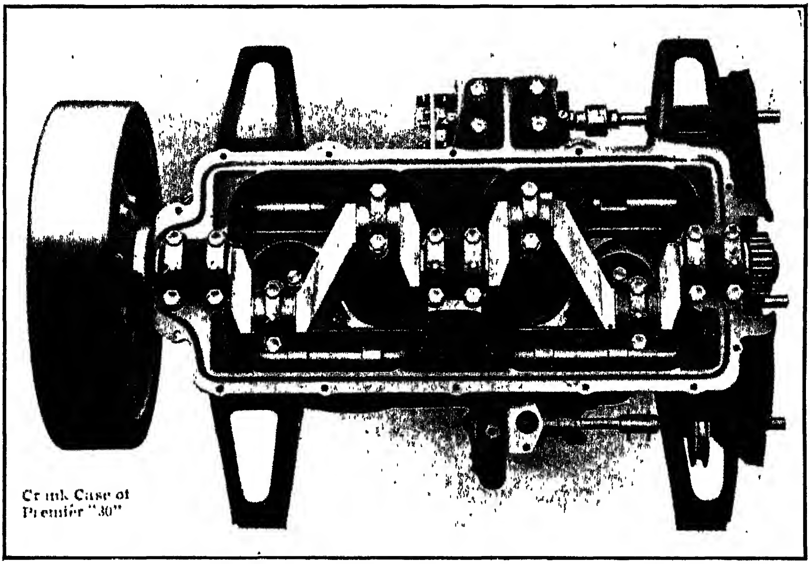


FIG. 149*a*.

mission corresponds with periods when both main valves are closed. When compressed air is turned on to the starting-valves, they will remain closed, excepting when the cam-throws are in such position as to engage with the valve-stems and force the valves open, each one in turn. The engine is, thus, conveniently started, from any position, simply by turning on the compressed air.

The Automobile Engine.—A modern four-cylinder type of automobile motor of high power is illustrated in Figs. 149*a* and 149*b*. The design is of the Premier Motor M'f'g Co. of Indianapolis,

Ind., and it represents the latest improvements in the American and European automobile motor construction.

Fig. 149*a* is an inverted plan view of the engine, with the lower part of the casing removed, so as to show plainly the arrangement of the crank-journals and the location of the cam-shafts which actuate the admission and discharge valves. The cam-shafts

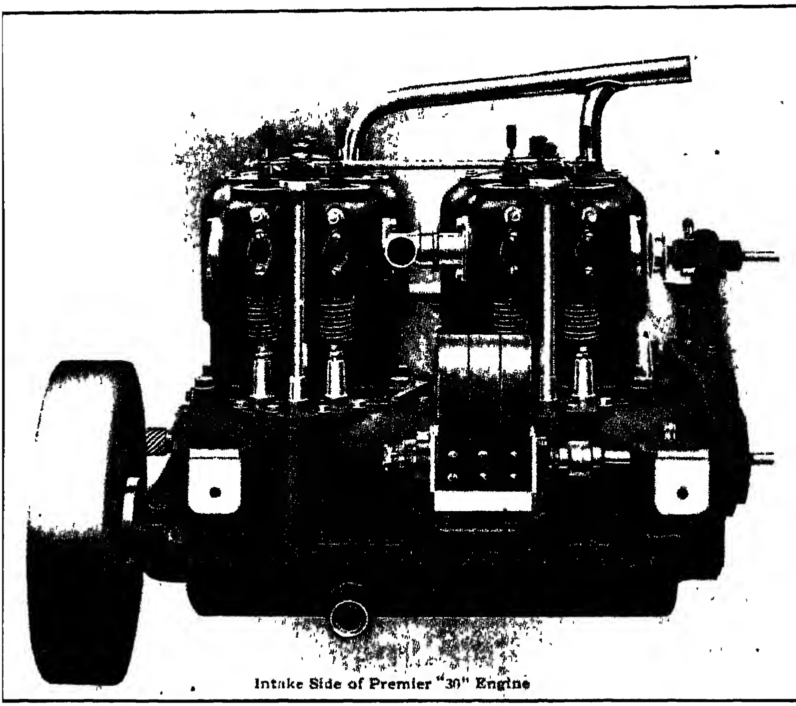


Fig. 149*b*.

are driven, it will be seen, from the end of the crank-shaft by means of spur-gears, which are arranged in a general gear-box at the front end of the engine. In the same gear-box are gears provided also for driving the circulating pump and the ignition-magneto. The igniter-spindles are operated from the admission cam-shaft by means of two pairs of spiral gears, while the lubricator is driven by means of spiral gears from the discharge cam-shaft.

Fig. 149*b* is a view looking from the admission side of the engine. It shows the arrangement of the four cylinders with their admission valve-casings, and the igniter-spindles coming up from the crank-case; each spindle operating the igniters for two cylinders. The magneto, of the Bosch type, is secured to the crank-case, in front of the cylinders, and its armature-spindle is driven, through a universal coupling, by means of a spur-gear in the general gear-box.

Figs. 150*a* and 150*b*, respectively a combined half longitudinal section and half front elevation, and a cross-section through one cylinder and crank-casing, show the detailed construction of the engines.

The cylinders, of a bore of 4½ inches, are cast in pairs, with the inlet and exhaust valve-chambers arranged on opposite sides of them. They are water-cooled as far down as the working space, and both valve-chambers are water-cooled, completely, all around the valve-seats. The cylinders and pistons are, of course, ground true, and the end of the piston is given a very slight taper, to allow for the expansion of the outside material when heated. Four eccentric piston spring-rings are used. They are ground true, and each pair fitted, freely, in each of the two grooves turned and ground near the end of the piston. Several oil grooves are, properly, cut in the lower part of the piston to retain and distribute the lubricant.

The crank-case, of close-grained cast iron, is made as light as possible, and the crank-case cover, to save weight, is made of sheet steel, principally. Aluminum cases have been used to some advantage, as far as the reducing of weight is concerned, but this metal does hardly seem to possess the necessary strength and stiffness to give the same rigidity to the framework of the engine as cast iron, and so far as durability of machined surfaces and threads are concerned it is considerably the inferior.

The crank-shaft is made of suitable drop-forged steel, and it is journaled in Parsons white-brass bearing shells, which metal, although it is of considerable stiffness, possesses the necessary softness for making a suitable journal.

The connecting-rods are made as light as possible by having

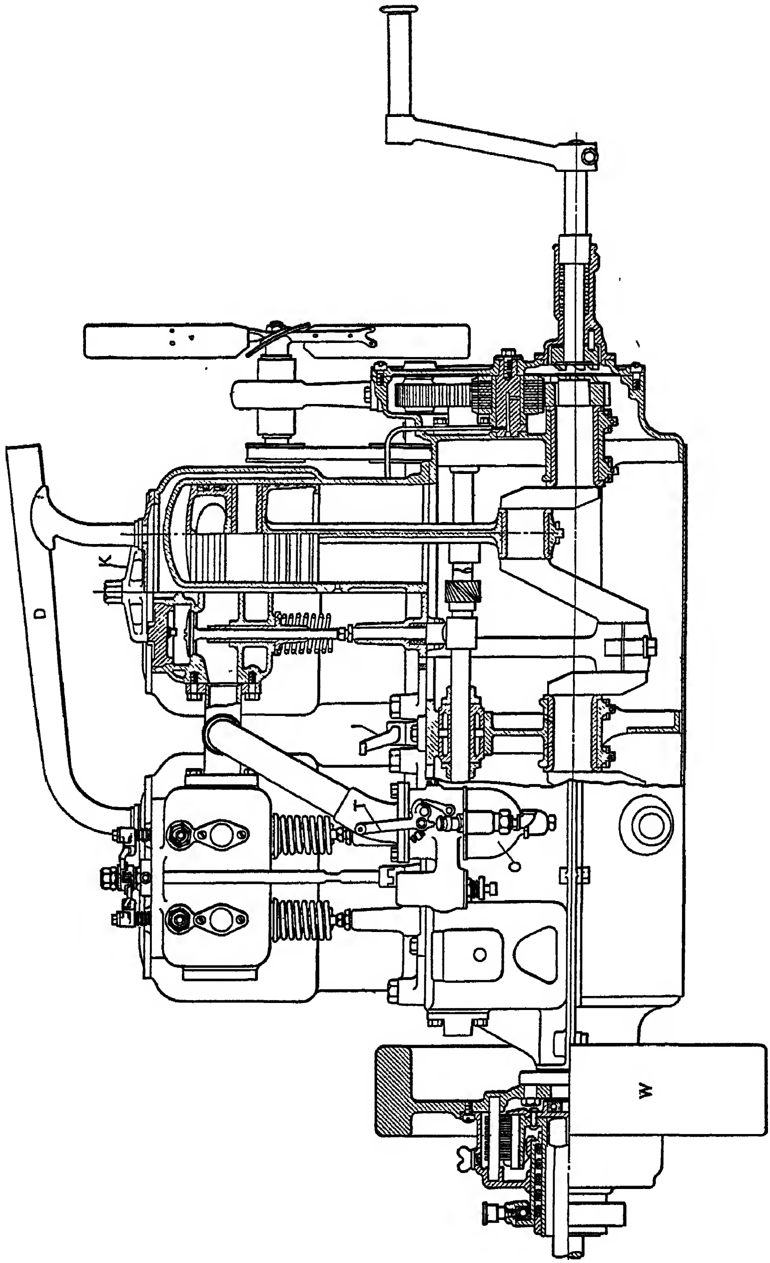


FIG. 150a.—Automobile Engine. Elevation and Longitudinal Section.

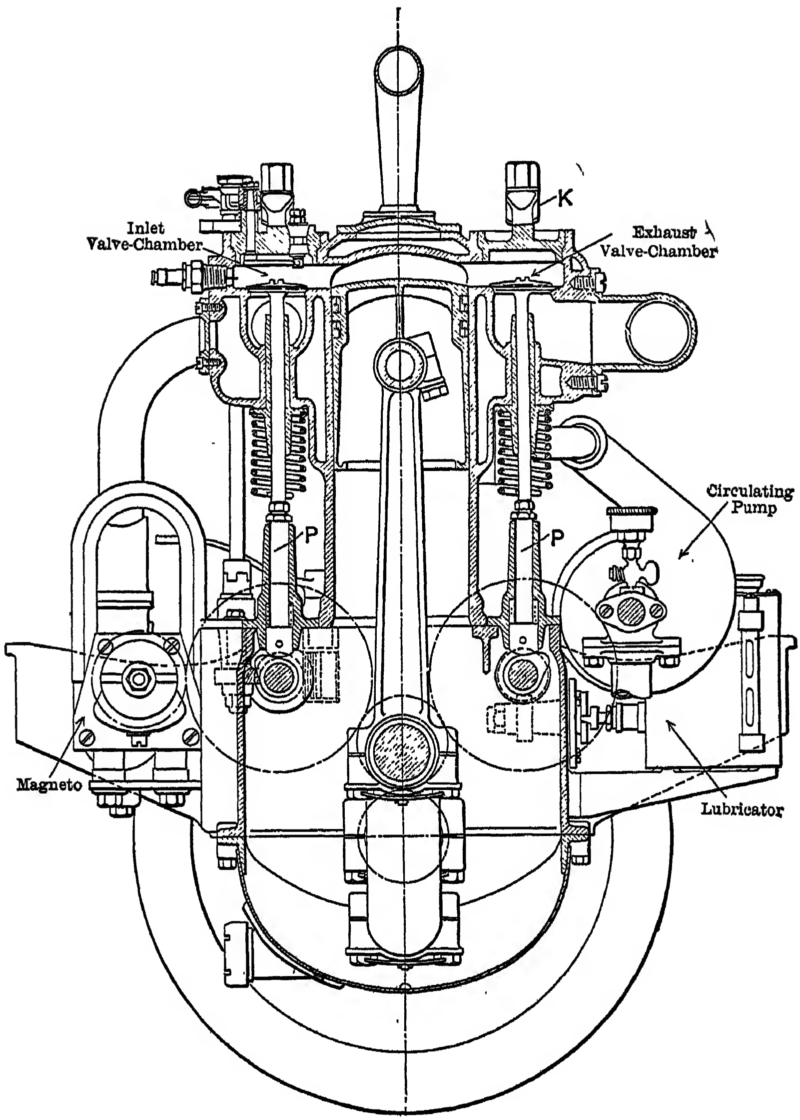


FIG. 150*b*.—Automobile Engine. Cross-Section.

the body of the rod formed in a channel-section; the metal, thus, well distributed for obtaining the greatest stiffness of the rod.

The valves are of nickel steel, and guided by means of their stems in long bushings so that they must always seat true. To effect the lifting of the valves, the push-rods, *P*, simply butt against the ends of the valve-stems, and the timing of the lift may, to some extent, be adjusted, by means of a locked adjusting screw in the end of the push-rod. Each pair of covers for the valve-casings are held closed by means of a clamp, *K*, which is tightened down on the covers by a heavy bolt. Thus, by unfastening four bolts the eight valve-covers become free for removal. In the covers of the admission valve-casings, as housings, there are arranged the igniter-rods of the low-tension make-and-break igniters. These covers serve, therefore, also the purpose of ordinary make-and-break spark-plugs, while high-tension spark-plugs are screwed into the side of the admission valve-casings. The two systems of ignition are applied to the engine, so that in case of failure of one the other can be resorted to.

The carbureter, shown at *C*, Fig. 150*a*, is equipped with a throttle valve, by means of which, in combination with the timing of the spark, the speed and power of the motor is controlled. The throttle-lever, *T*, and the igniter-lever, *I*, are controlled by means of throttle- and spark-levers in the steering wheel. The latter are adjusted over a stationary quadrant in the wheel.

The cooling system employed in connection with these engines is in principle the same as that described at page 328. The hot water discharge pipe from the top of the cylinders to the radiator is shown at *D*.

A fan for impelling an air-current through the cells of the radiator, which is placed directly in front of the engine, is shown in Fig. 150*a*, and it is driven by means of a belt from the circulating-pump shaft.

A forced-feed lubricator is located on the exhaust side of the motor, in close proximity to the exhaust-pipe, which thus keeps the lubricant, at all times, at an even and suitable temperature for efficient lubrication. The lubricator is, as has been explained, driven by means of a pair of spiral gears from the discharge cam-

shaft. The oil is fed to the cylinder at a point between the upper and lower set of spring-rings when the piston is at its lower centre.

The make-and-break igniter mechanism, shown for one pair of cylinders in Fig. 149*c*, consists of two small cams secured to the end of the igniter spindle; actuating, each, one of the two igniter cam-levers. The cam-levers are not secured directly to the movable electrodes, but the latter, which are held closed independently by springs are actuated through a forked arm. The cam-levers have ample clearance between the lugs of the

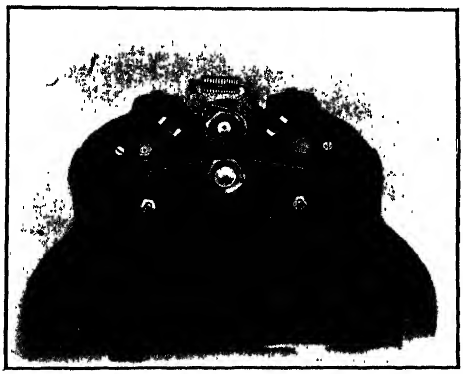


FIG. 149*c*.

fork to allow the igniter to close positively and to be free of any vibrations that may be set up in the levers.

A Weston type multiple-disc clutch, *W*, is employed for clutching in or releasing the motor from the transmission shaft. This clutch consists of a housing attached to the fly-wheel of the motor and of a hub keyed to the transmission shaft. Between these two there are a series of disks, every second of which is fastened to the housing and every second to the hub. By means of a sliding collar, the disks can be squeezed together, tightly, which, due to the friction between the various disks, will cause the transmission shaft and the motor-shaft to revolve together. By varying the pressure with which the disks are pressed together the latter will slip relatively to each other, more or less, thus

varying the speed of the transmission shaft. The clutching in of the motor, by means of this clutch, will always be gradual, and, thus, any sudden shocks in the shafts and gearing are avoided.

The engine is secured solidly to the frame of the vehicle by means of brackets seen, in Figs. 149*a* and *b*, to extend from the crank-case, and in front of it is installed the radiator, which forms the front end of the bonnet enclosing the motor; as seen in Fig. 151. The latter figure shows also the various controlling levers, and those for shifting the clutch-collar and transmission-gears. The figure illustrates, besides, in a striking manner, the relatively small space occupied by a 45-horse-power motor, together with its auxiliaries, as installed in a touring car.

Kerosene and Oil Engines.—The Hornsby-Akroyd Oil Engine— In the Hornsby-Akroyd oil engine, shown in a longitudinal section in Fig. 152, the air only is admitted to the cylinder during the suction stroke, and the fuel-oil is by means of the oil-pump, introduced into the vaporization- and combustion-chamber, *V*, at the time the engine piston commences to draw its air. In this chamber, which is, as seen, separated from the cylinder by a contracted passage, *P*, the fuel is vaporized and mixed with the neutrals remaining from the preceding stroke. When the air-charge, which in the meantime is being compressed in the cylinder, finally, toward the end of the compression stroke, is forced into the vaporization-chamber and mixed with the fuel, the charge will ignite due to the heat of the vaporization-chamber. The expansion stroke and the discharge of the burned gases follow during the next forward- and return-strokes. The engine operates, thus, according to the Otto four-stroke cycle, and with self-ignition.

The air-inlet and exhaust valve-chambers are located side by side back of the cylinder, as viewed in Fig. 152, and stand in communication with the cylinder through the port *A*. The valves are operated by means of the valve-rockers, rollers and cams seen below and in the front of the cylinder, in Fig. 153.

From the latter illustration a fair idea of the general construction of the engine and of many of its details can be obtained.

The oil-pump is operated from the air-inlet valve-lever; the

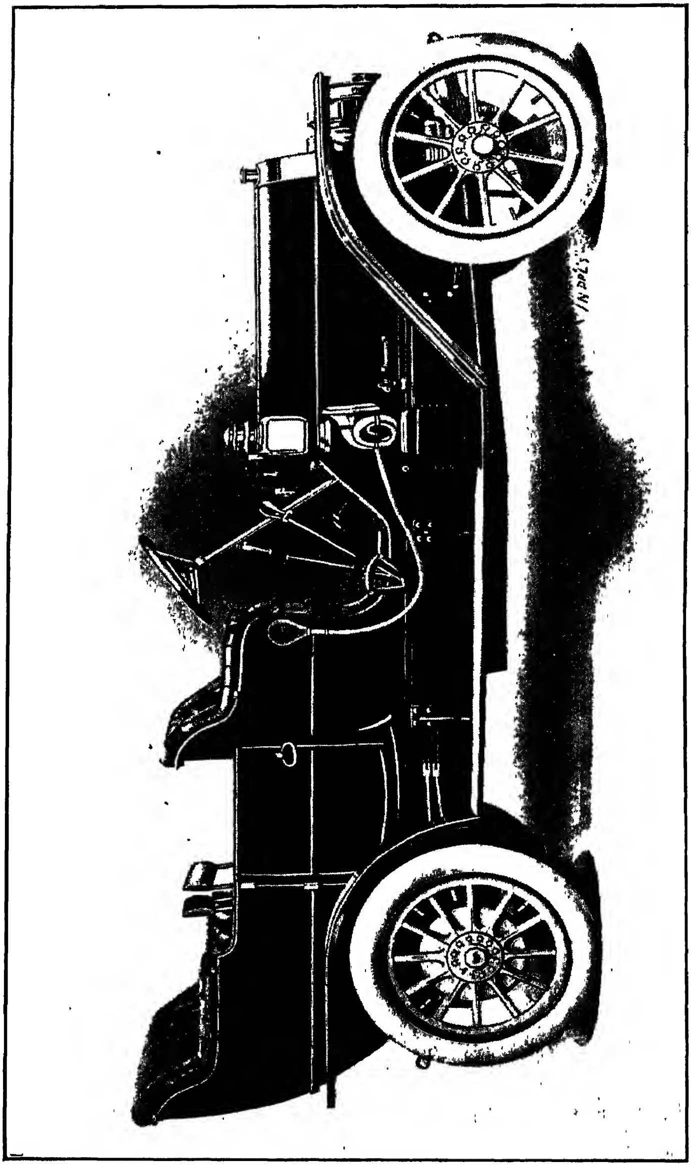


Fig. 151.

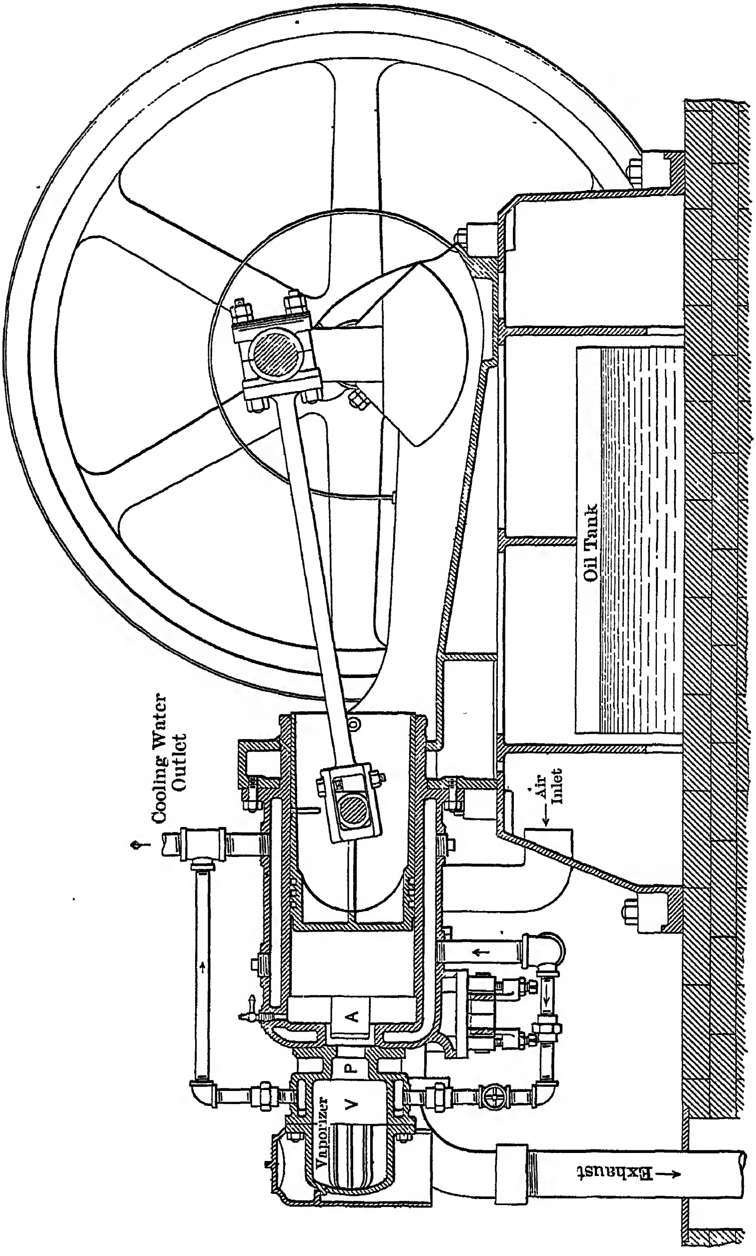


FIG. 152.—Section through Hornsby-Akroyd Oil Engine.

fuel, thus, being forced in to the vaporizer at the time the air valve is being opened for the admission of air to the cylinder, and the amount of oil introduced is controlled by the governor.

Before the engine can be started, the vaporization-chamber must be heated to the required temperature to cause the ignition of the first charge. For the purpose, there is used a torch which is applied underneath the vaporizer and inside the hood which forms the protection for the latter.

The Diesel Oil Engine.—The Diesel engine operates, as has

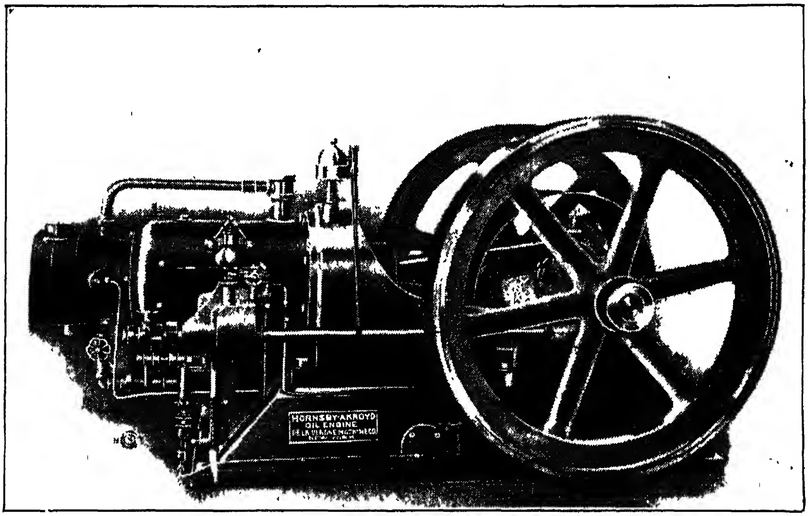


FIG. 153.—Hornsby-Akroyd Oil Engine.

been explained in the preceding, page 24, according to the Brayton or Diesel cycle. It is built commonly in units of one, two-, three-, or four-cylinder machines and of powers of from 25 to 400 brake horse-power.

The engine operates on any petroleum fuel, crude or refined, whose gravity is not less than 19° Baumé. Fuels of 30° Baumé gravity, with a flash-point of 140° to 240° F., are particularly suitable.

Fig. 155 is a cross-section through one cylinder and the crank-case, and it shows plainly the detailed construction of the engine.

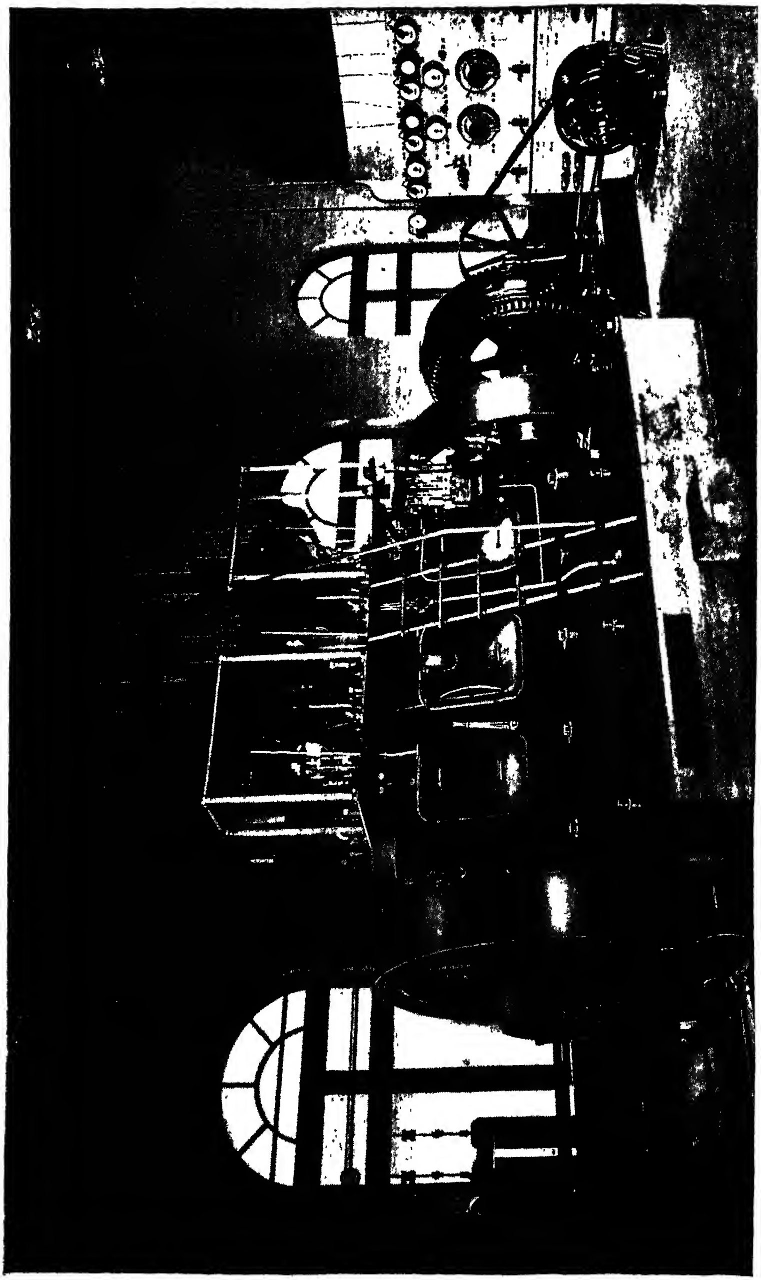


FIG. 154.—Installation of Diesel Three-Cylinder Engine.

The air-inlet valve, the exhaust valve, and the fuel valve are as noted in the figure. Each of the above valves is driven by separate cams and cam-levers located inside the crank-casing, and the timing of the valves is according to the diagram Fig. 156.

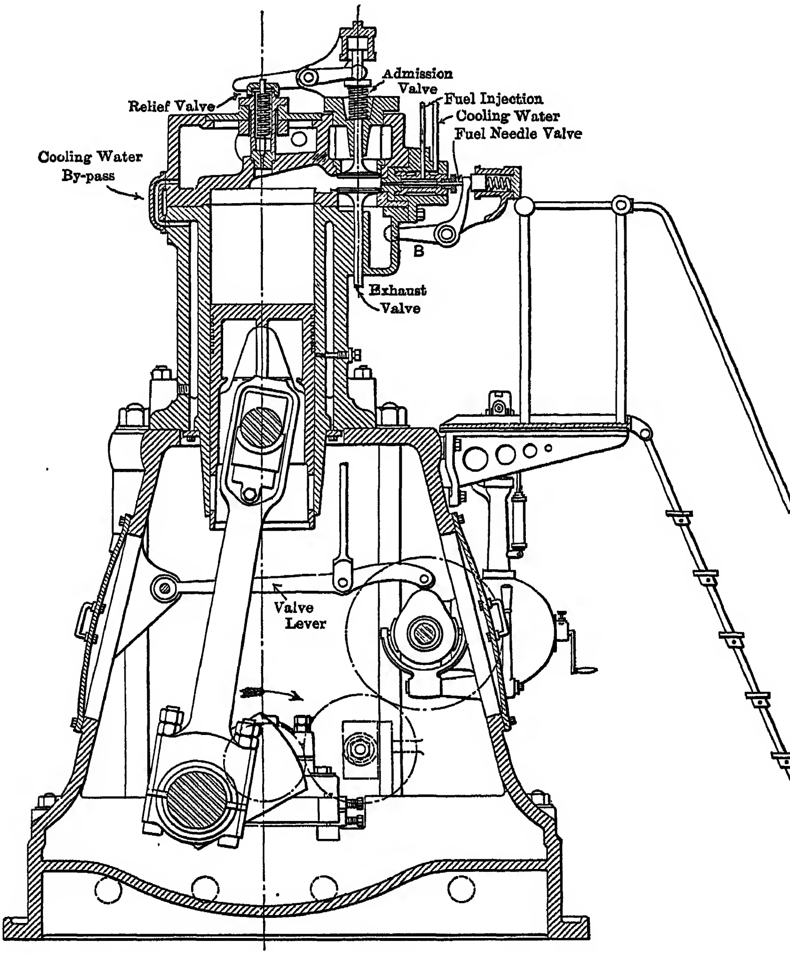


FIG. 155.—Transverse Section through Diesel Oil Engine.

The fuel valve consists of a needle-valve which seats against a steel bushing provided with an admission port of suitable size, through which the liquid fuel-oil is injected by means of an

atomizing jet of air under high pressure. To the fuel-valve casing there is connected, as shown in the figure, the fuel-injection pipe and a water pipe for the cooling of the fuel-valve.

If the operation of the engine be followed through one cycle (consisting of four strokes), it will be as follows: The cycle begins with the suction stroke and the piston on the top centre, and while the piston descends pure air will be admitted to the cylinder, which on the upward stroke will be compressed into the combustion-space to a pressure of approximately 500 pounds. The adiabatic compression of air to 500 pounds will cause its temperature to rise from, say, 60° F. to 930° (computed according to formula 33*a*). Practically, due to absorbed heat, the temperature will become close to 1,100° F., which, of course, is high enough for igniting spontaneously any crude or refined petroleum that may be admitted into the air-volume. When the piston arrives at the top of its compression-stroke the bell-crank B, will open the fuel valve, and the fuel is injected, becomes ignited, and burns. The amount of fuel that will be injected is regulated by the governor which is in immediate control of the fuel pump supplying the fuel-charge to the injection pipe, and the supply will, accordingly, be proportionate to the load the engine carries.

After the closing of the fuel value, which occurs when the piston has travelled approximately eight per cent of its downward working stroke, and the fuel is consumed, expansion of the enclosed gases will continue, until, at 90 per cent of the stroke, the exhaust valve opens for release. During the following upward stroke the gases are discharged from the cylinder.

The air for the injection of the fuel is compressed by means of an independent air-compressor to about 1,100 pounds, and it is cooled in a special air tank before it is used.

An installation of a three-cylinder Diesel engine, connected to an electric generator, is represented in Fig. 154. Toward the extreme left of the figure the storage tanks for the compressed injection-air are shown, and the compressor that furnishes the air to the tanks is seen to be belted from the left-hand pulley of the engine.

At full speed of the engine the air charge drawn in to the

cylinder will necessarily be of a pressure somewhat less than the atmospheric, due to resistances in valves and ports; and the charge is compressed to approximately 500 pounds. At a slow speed, in starting the engine, on the other hand, the charge will be of a pressure of practically that of the atmosphere. It has been pointed out that, for high compression, the final pressure of the charge will be considerably higher when the compression begins at the atmosphere than it will when the compression begins only a trifle below the atmosphere, and, in the case of the Diesel engine, the increase in the final pressure at slow speed is so considerable that a relief valve must be employed to release it. This valve, commonly set to lift at a pressure of 800 pounds, is shown in the sectional view of the engine, Fig. 155.

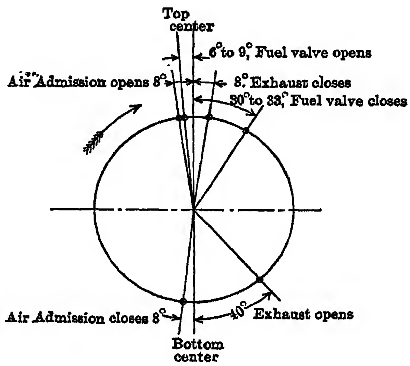


FIG. 156.—Valve-Cam Setting for Diesel Oil Engine.

The Diesel type of engine has, lately, been successfully used for marine purposes. The safety of its fuel and many features about the engine make it seem particularly suitable for such service.

The Double-Acting Four-Cycle Engine.—In this country, with respect to large power-requirements, the double-acting four-cycle engine has lately become the standard engine.

The distribution of the power during the cycle is not in the double-acting single-cylinder engine as favorable as in the tandem or twin single-acting, and it is therefore rarely built as a single unit. Its power-capacity, for the space occupied, is, however, much greater, and on this account the application of the single-acting engine will in the future undoubtedly be limited to power-requirements of approximately 300 to 400 horse-power; and above that power the field for the double-acting tandem, or twin-tandem, will begin.

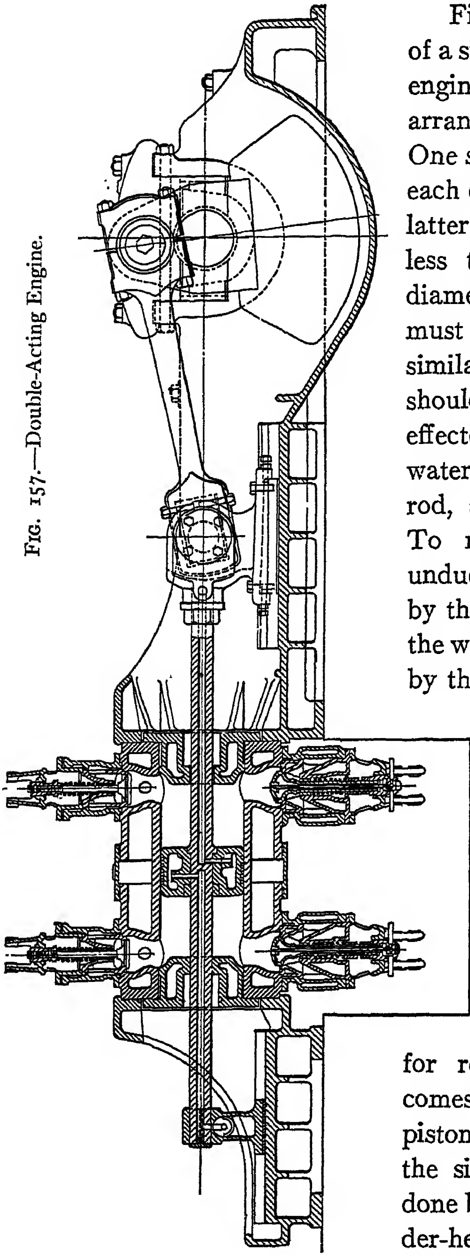


Fig. 157.—Double-Acting Engine.

Fig. 157 is a sectional view of a single-cylinder double-acting engine, which shows the general arrangement of its main details. One set of valves is provided for each end of the cylinder, and, the latter not being generally made less than 20 to 22 inches in diameter, the exhaust valves must be water-cooled. For a similar reason the piston also should be cooled, and this is effected by leading the cooling-water through a hollow piston-rod, as explained at page 296. To relieve the cylinder from undue wear liable to be caused by the dead load of the piston, the weight of the latter is carried by the main and the rear cross-heads.

The cylinder-heads are thoroughly cooled and provided with suitable metallic rod-packings, which often are, as in the Schwabe packing, separately cooled.

The piston is secured to the rod solidly, not for removal, and when it becomes necessary to remove the piston from the cylinder it is, in the single-cylinder engine, best done by removing the rear cylinder-head and sliding the piston and rod together out through the

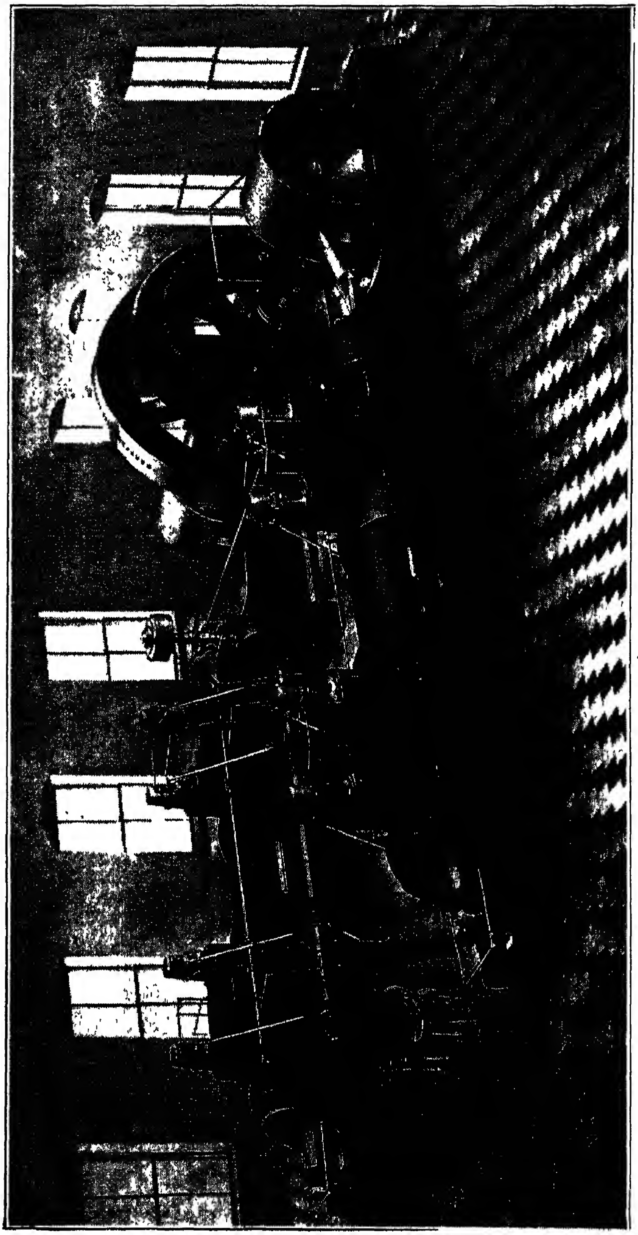


FIG. 158.—Allis-Chalmers Twin Double-Acting Tandem Engine. Direct-Connected to Electric Generator.

rear end of the cylinder. To facilitate this, ample space must be provided at the rear end of the engine. The front piston of a double-acting tandem engine must be removed through the front end of the cylinder, and this is accomplished by removing the main crosshead and front head, and, with the crank and connecting-rod in such a position as not to interfere with the piston-rod, the piston may be slid through to the front.

A double-acting tandem engine, direct-connected to a generator is represented in a plan view and in elevation, in Figs. 159 and 160. It is essentially the same engine as the single-cylinder engine, Fig. 157, with the addition of the rear cylinder and an intermediate crosshead and guide. The piston-rods are coupled together at the intermediate crosshead, and the rods and pistons are removed, one through the front and one through the rear end of the front and rear cylinders.

The cooling-water for the pistons

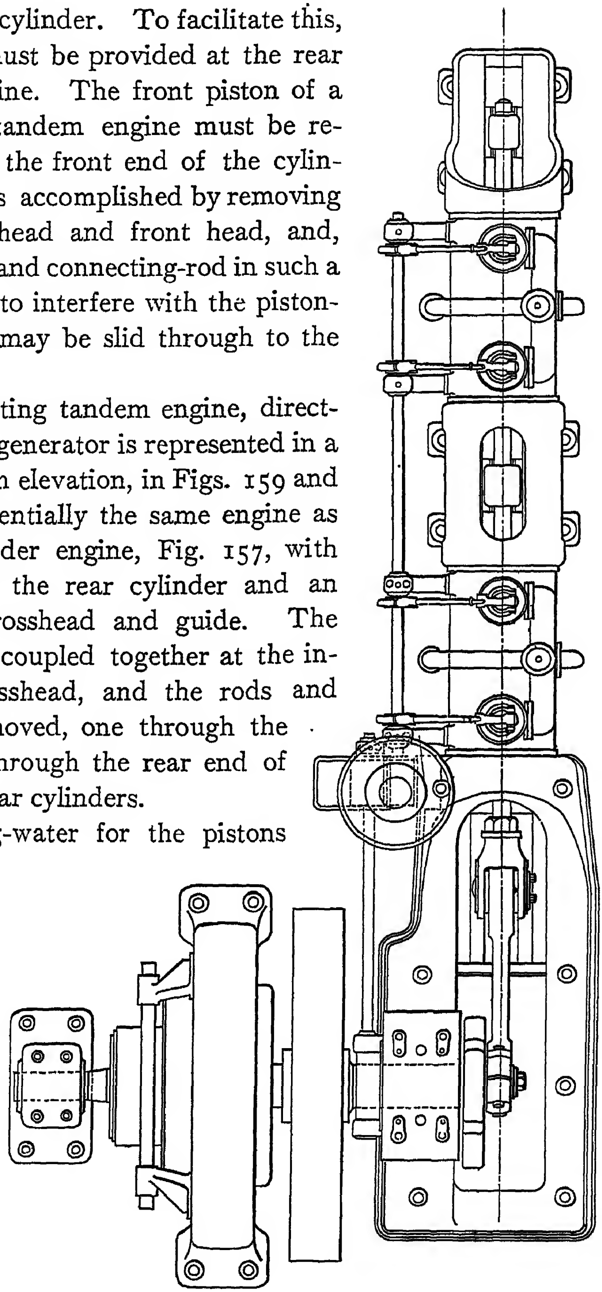


FIG. 159.—Four-Cycle Double-Acting Tandem Gas-Engine. Direct-Connected to Electric Generator. Plan View.

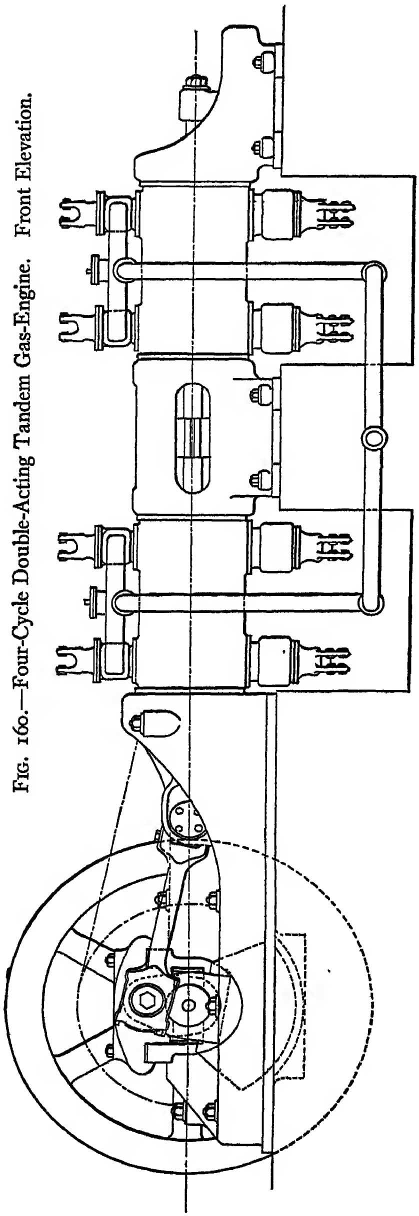


FIG. 160.—Four-Cycle Double-Acting Tandem Gas-Engine. Front Elevation.

is sometimes admitted at the intermediate crosshead, for both pistons; one stream going to the front, to be drained from the main crosshead, while the other goes to the rear, to be drained from the rear crosshead. Sometimes the cooling-water is admitted at the main crosshead, flowing through one after the other of the cylinders and drained from the rear crosshead. The latter arrangement will cause the water going through the rear piston to be somewhat hotter than that going through the front one; and this, it is claimed, will be of disadvantage. The temperature-difference between the gases and the water is, however, so considerable that the matter of some degrees higher or lower temperature of the cooling-water will be of minor importance.

Whenever there are sulphurous vapors present in the gas, the water should be run through the rods hot, so as to prevent any water-vapor from condensing in the rod packing-boxes. Any moisture will, namely, have for effect to absorb the acids formed by the sulphur in the gas, and cause corrosion of the metallic parts.

The Allis-Chalmers Engine.—Lately, large double-acting tandem engines have frequently been installed to work on natural gas, producer gas, or blast-furnace gas. Fig. 158 is a general view of a double-acting twin-tandem four-cycle engine built by the Allis-Chalmers Co. of Milwaukee, Wis.

A cross-section through the cylinder and valve-casings of the Allis-Chalmers engine is shown in Fig. 161, from which the construction of the valve-actuating mechanism may readily be understood.

The valves, it will be seen, are driven from a lay-shaft by means of eccentrics and rolling levers; the latter having for object to effect a quick opening and closing of the valve-ports, though the initial lifting and the seating of the valve is performed only very gradually.

Referring to the set of rolling levers R and F, which actuate the exhaust valve and which are fulcrumed at r and f, it will be seen that when the eccentric E begins to pull the lever-pin p upward, tending to open the valve, the leverage by which the valve is actuated upon is large, at first, and hence the motion of the valve, in starting, slow. As the lever-pin p is pulled higher, however, the total leverage decreases gradually, and hence, the speed with which the valve is lifted grows quicker the higher the valve is raised. At the closing of the valve the reversed motion will, of course, obtain. The valve, thus, starting to close quickly, will seat only very gradually. When seated, the valve is held closed by a sufficiently heavy spring and the motion of the rolling levers becomes free and independent of the valve.

The exhaust valve is placed at the bottom side of the cylinder, which is often considered the most suitable arrangement, as it will allow the combustion-chamber to free itself most readily from oil, or impurities that may come with the gas. As generally required in engines above 20 inches, the exhaust valve is water-cooled and its construction is practically identical to that of the water-cooled valve illustrated and described at page 296. The cooling water is supplied by means of a flexible hose connected at W and the discharge water is carried off by the same means from the opposite side of the valve-stem.

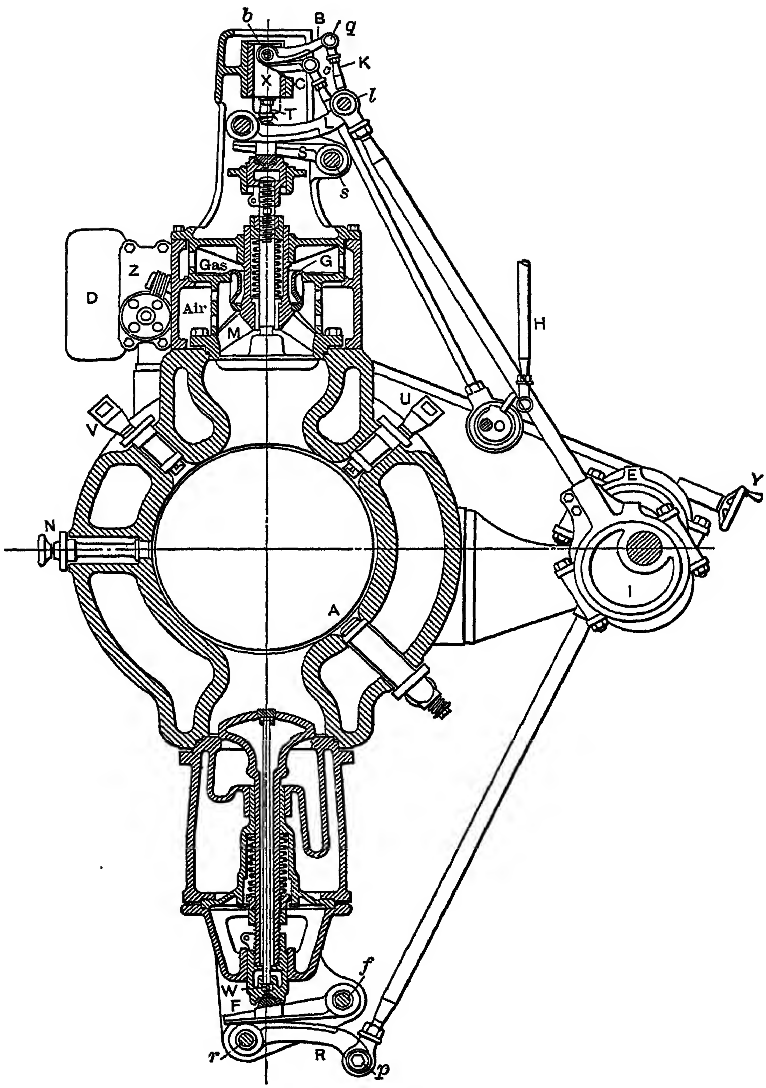


FIG. 161.—Allis-Chalmers Valve Gear.

The gas-valve, G, is double-ported and surrounds the main valve-stem. It is actuated from the small crosshead X with which it is connected by means of two tie-rods T. The gas and air are supplied separately through the manifold D to the gas- and air-chambers above and below the gas-valve. The timing of the main inlet valve and gas-valve is such that the main valve opens more or less in advance of the gas-valve at the beginning of the suction-stroke. Thus, at first, air only is admitted to the mixing-chamber and cylinder, and consequently any tendency to back-fire, on account of any lingering flame, or sparks, in the cylinder when the inlet valve opens, will be minimized. After the gas-valve has become opened the mixing of the fuel and air will take place in the mixing-chamber M; the gas which is supplied through the gas valve-ports meeting currents of air entering through numerous holes in the walls of the mixing-chamber, in a manner very favorable for effecting an intimate mixture between the two gases. The supply of a suitable fuel-mixture to the cylinder will continue until the gas-valve closes, always a little ahead of the closing of the main inlet valve. The object in delaying the closing of the air-supply being to scavenge the mixing-chamber by pure air from any explosive mixture.

The operation of the valve gear for the inlet valve will be as follows:

The main inlet valve is actuated by means of the inlet eccentric I and the rolling levers L and S, and the motion of the gas-valve will be the same as that of the small crosshead X with which it is connected. On the crosshead X there is pivoted the rolling lever B by means of the pin b, while the fulcum lever C, on which the former rolls, is forked around the crosshead guide, and pivoted on the valve-bonnet. The lever-pin q from which the gas-valve is operated and the lever-pin l from which the main valve is operated are connected by means of the link K, and, hence, the two levers will be moved in unison by the inlet eccentric.

The fulcrum lever, S, for the main valve being solidly hinged on the valve-bonnet, at s, the motion of the inlet valve will be uniformly the same for every cycle. The position of the free end c of the fulcrum-lever for the gas-valve is, however, under control

of the governor, in such a manner that a small eccentric O on the governor-shaft will raise or lower the end of the fulcrum-lever according to the position of the governor. The timing of the opening of the gas-valve, and the lift of the valve, will thus depend on the position of the governor, and will be varied according to the load on the engine. When the governor rises the position of the fulcrum-lever C will drop, the opening of the gas-valve will occur later, and its lift will be decreased; and when the governor rises the opposite in each respect will be the occurrence.

The eccentric O may be disconnected from the governor-shaft, and swung, by hand, to its lowest position, which will in effect prevent the gas-valve from opening at all. This manipulation becomes of use whenever it may be desired to cut out any one of the combustion-chambers. The rocking motion of the governor-shaft is controlled by means of the connection H, whose upper end attaches to the governor-lever.

As is usual with respect to large engines a double set of igniters is provided, located at U and V. The practice of duplicating the means for igniting the charge has become general, on account of the liability of one set failing, often from a very slight cause.

At N is shown a valve-spindle which closes the indicator-opening. By removing this valve-spindle from the outside end of the indicator bushing a thread will be exposed for the attachment of a standard indicator-cock.

At Z is shown a valve for the proportioning of the air and the fuel in the charge. The adjustment is made from the operating side of the engine by turning the hand-wheel Y. It may become necessary to re-adjust the setting of this valve occasionally during a run, as few fuel-gases in general use are so constant as not to vary quite materially, in heating-value and composition, from time to time.

A check-valve for admitting compressed air for the starting of the engine is provided at A. The valve is seated on the inside end of the valve-chamber, and is held closed by a spring acting on the prolonged valve-stem which runs through the chamber. At the outside of the valve-chamber, close to the cylinder, means for the attachment of the compressed-air piping is provided.

The Nuernberg Engine.—A well-known German engine of the same general design as the one just described is the Nuernberg double-acting engine, the latter being, in fact, the original design from which the Allis-Chalmers engine has been developed. A test of the Nuernberg engine on blast-furnace gas is recorded in Table XXXI, pages 410 and 411.

The Westinghouse Engine.—The general construction of the Westinghouse double-acting engine is clearly shown by the half-tone reproduction of a twin-tandem engine, Fig. 162. One of the features of this engine which differs from the Allis-Chalmers-Nuernberg practice is that the outer cylinder-jacket wall is not made continuous from end to end of the cylinder, but is cut circumferentially by the jacket core at the middle of the cylinder, the object being to provide means for the free expansion and contraction of the jacket wall, so as to avoid strains due to the unequal heating of the inner cylinder barrel and the outer wall. The opening in the outer wall is closed by a cast-iron belt which is clamped tightly around the cylinder so as to form the water-jacket.

The governing of the engine is effected by means of a combination throttling and cut-off regulation; the charge of an approximately constant-quality mixture being throttled by the governor to suit the load, and cut off by the closing of the mixing valve.

Referring to Fig. 163, which represents a cross-section through the cylinder and valve casings, it will be seen that the inlet and the exhaust valves are both operated by the same eccentric. The exhaust valve, as is necessary in large engines, is water-cooled; the cooling-water being supplied, and the discharge-water carried off, by flexible hose-connections. The motion of the exhaust valve is transmitted from the eccentric through a pair of rolling levers, F and R, which give the valve, in lifting it, a rapidly increasing speed, yet starting it from its seat and seating it only very gradually.

The pressure on the valve, when being opened, is approximately 30 to 40 pounds per square inch and, hence, it becomes quite necessary that the first cracking of the valve-opening is performed as gently as possible, to avoid severe strains and jar

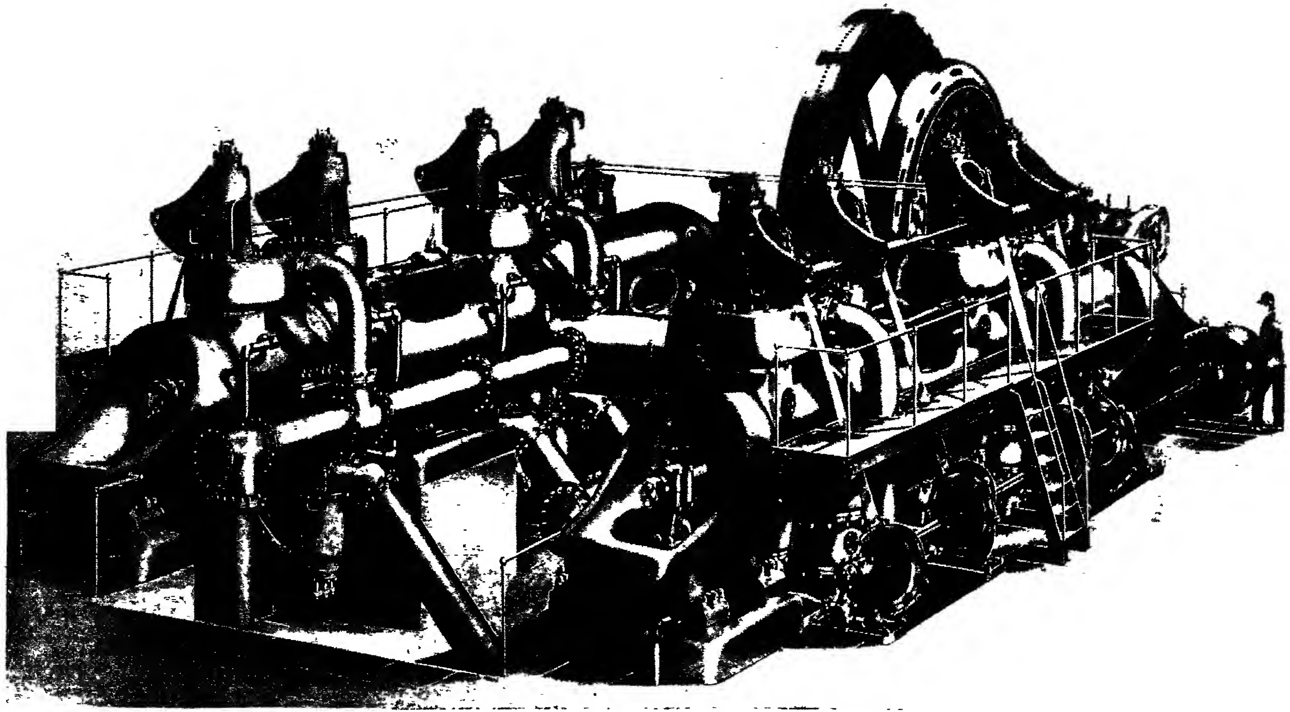


FIG. 162.—Westinghouse Double-Acting Twin-Tandem Engine.

in the valve mechanism. After the valve has become opened, however, there is only the spring-tension to resist its motion and it becomes then desirable to increase the speed with which it is raised, so as to avoid as much as possible the wire-drawing of the discharge.

The admission and mixing valves are actuated by means of the connection *N* and the rolling levers *L* and *K*. The mixing-valve *M* consists, as is seen in the figure, of a cylindrical sleeve mounted on the main admission-valve stem, in such a manner that it follows the motion of the latter up and down, but it is free to rotate about the valve-stem under the influence of the governor. In this sleeve there are provided, for the admission of the gas and air, two sets of rectangular port-openings *o* and *p* which register, when the main valve is fully open and the governor is down, with corresponding ports in the sleeve casing. Between the port-openings of the same set in the sleeve there are bridges somewhat wider than the openings, and if the governor is raised to its highest position the valve-sleeve will be rotated to such a position that when the main valve becomes opened the bridges between the ports in the sleeve will cover the ports in the valve casing and exclude entirely any charge from the cylinder. Between these positions of the mixing-valve, that for the admission of a full charge of air and gas and that for not admitting any charge at all, the governor will have control to rotate it to a position to suit the requirements of the load.

The main admission valve is in the figure shown in its closed position, and it will be noticed that the air-port is covered by a lap somewhat less than that of the gas-port. This will have for effect, that, when the main valve is closing, the gas will be cut off a little earlier than the air, and hence the mixing chamber will, after each suction-stroke, become scavenged by the last incoming air.

The governing of the Westinghouse multiple-cylinder engine of this type, which must be accomplished by revolving four or more mixing-valve sleeves to a position to suit the momentary load, with certainty, is effected indirectly by means of hydraulic power. This becomes necessary, as the valve-sleeves are liable

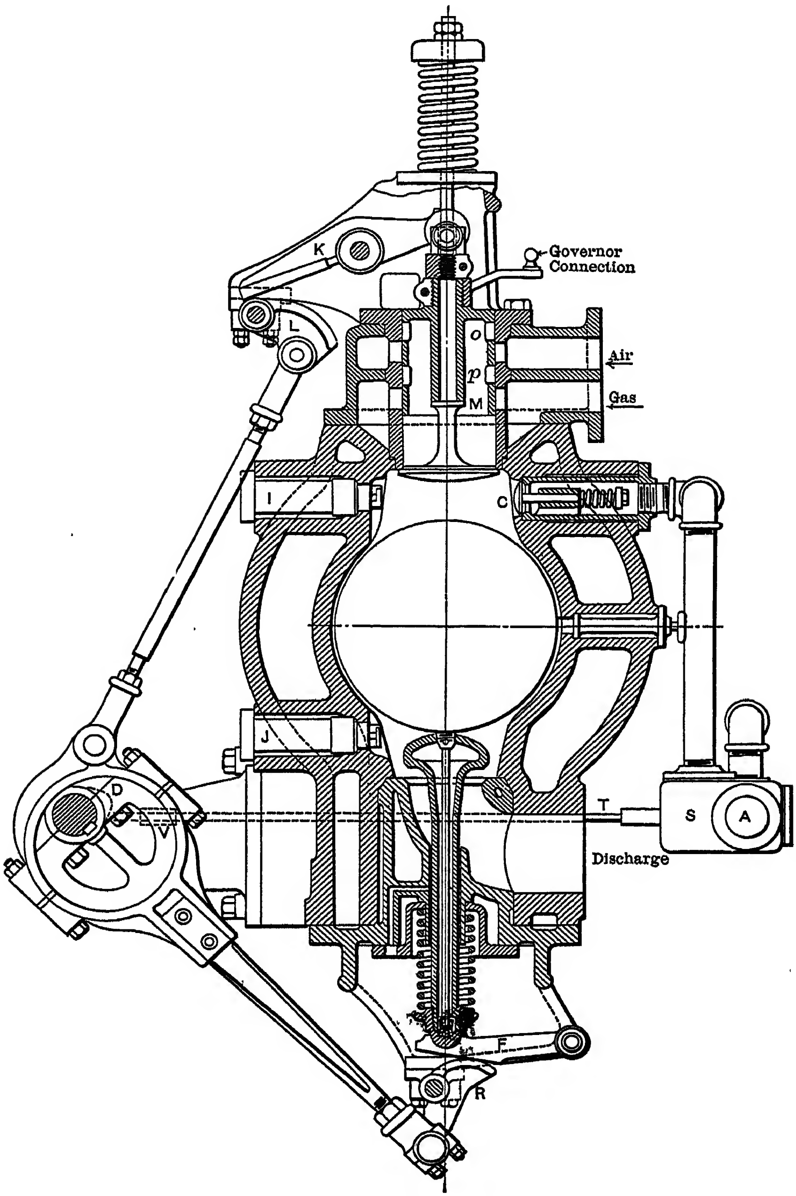


FIG. 163.—Westinghouse Valve Gear.

to stick, due to collected impurities. The governor proper is of the Hartung type and controls the valve of a small hydraulic cylinder from which the power for adjusting the mixing-valve sleeves is obtained.

The indirect method of governing is described, in detail, at page 314. For an engine of large power the complication involved by this method of governing is of minor importance.

Air-Starting Arrangement.—The starting, with compressed air, of a gas-engine having one combustion-chamber for each stroke of the complete cycle is a very simple matter, as in this case the engine may be started from any position it may have, excepting, of course, from the dead centres. The air-starting arrangement for all multiple-cylinder double-acting engines is in principle the same as that shown in connection with the Westinghouse valve gear, Fig. 163. It consists of a small spring-closing air valve for each combustion-chamber of the engine. This valve is located in the valve-chamber *S*, and its valve-stem, *T*, is prolonged so as to reach close to the small cam, *D*, on the valve-gear shaft, by which it is actuated for opening the valve. The valve-stem is, of course, guided near the cam, at *V*. The timing of the opening and closing of each of the small starting valves is such that air will be admitted to the combustion-chambers during the period of the regular expansion-stroke, when the main admission valve as well as the exhaust valve are closed.

C is a check-valve for admitting to the engine the compressed air for starting. The valve is seated on the inside end of the check-valve chamber, while the outside end of the chamber is connected with an air pipe leading to the starting valve *S*, which is supplied through the opening *A*.

As soon as air is turned on to the starting valves, each one of the combustion-chambers of the engine will in turn, receive air pressure, and the engine as a whole will work as a compressed-air engine, until the gas-supply valves are opened and the fuel-charge is fired regularly. The compressed air is generally not admitted to the cylinder before the crank has well passed the centre; hence, should the regular fuel-charge explode, usually, in starting, at the time the crank passes the centre, then the check-valve will remain

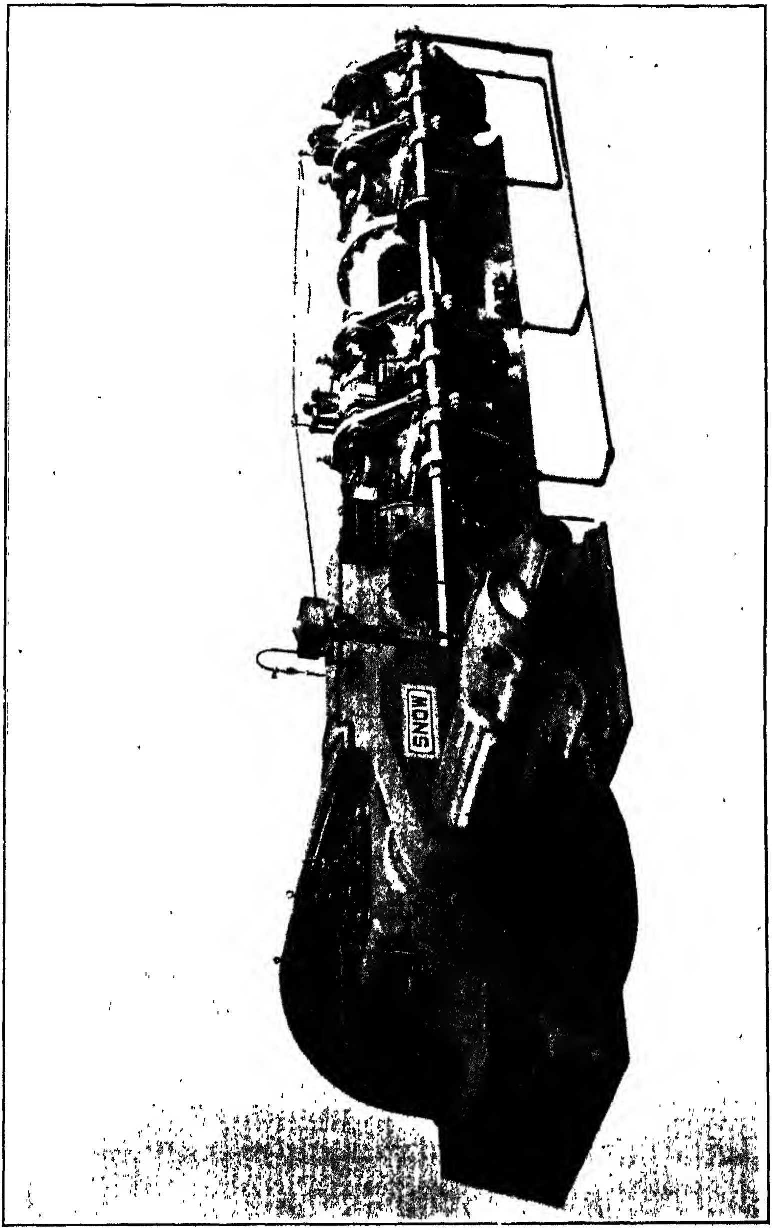


Fig. 164.—"Snow" 500 H. P. Double-Acting Tandem Engine.

closed against the compressed-air admission, due to the explosion-pressure in the cylinder.

The "Snow" Engine.—Fig. 164 illustrates a 22×36, 500 horse-power, double-acting tandem four-cycle engine running on illuminating gas, 130 revolutions per minute. It is installed at the People's Gas-Light and Coke Co., Chicago, Ill., and is built by the Snow Steam Pump Co. of Buffalo, N. Y. This engine, it will appear, differs from those of the same type just described in that the valve-chambers are here placed on the side of the cylinder instead of at the top and bottom; an arrangement which has the advantage of leaving the engine foundation continuous from end to end, without any breaks for the accommodation of the exhaust valves.

In Figs. 165 and 166 is illustrated a standard "Snow" valve gear; however, of somewhat different construction from that of the engine shown in Fig. 164. Fig. 165 is a cross-section through the valve-casing and Fig. 166 is a cross-section through the cylinder.

The valve gear is of a release gear type. The inlet and exhaust valves are arranged, the former above the latter, in a valve-chamber, *V*, which is placed at the front side of the cylinder. With respect to the exhaust valve there is no new feature of particular note. It is of a mushroom water-cooled type, and the discharge-water from the valves is carried by a discharge-hose to a jacket for the cooling of the exhaust pipe. A cam-shaft, *A*, running along the front of the engine, drives the valves.

Referring to Fig. 165, the main inlet valve, *I*, and the cut-off or mixing valve, *M*, are arranged side by side in the same valve-casing. The main valve is actuated directly by the valve-rocker, *R*, which is forked around the spring-seat so as to exert a normal force, by pressing on two diametrically opposite points, *a a*, of the lower flange of the spring-seat, when opening the valve. The closing of the valve is effected by a heavy spring, *S*. The upper end of the main valve-stem is connected, by means of a ball-and-socket connection, to one end of the mixing-valve rocker, *U*, the other forked end of which hinges to the crosshead *X*. The rocker, being fulcrumed by means of a pair of links *L*, transmits, thus,

to the crosshead a reciprocating motion in unison with that of the main valve. The crosshead *X* is guided in a bored guide in the upper part of the valve-bonnet, *B*, and it carries the hook arrangement by which the mixing-valve is lifted or released.

The detailed arrangement of the release gear is shown in Fig. 166. On the crosshead, *X*, there is hinged the hook *H*, in such a manner that the catch-block *N* will engage with the block, *O*, on the head of the valve-stem, when the inlet valve is closed. The spring *Z* exerts tension enough on the hook to cause the blocks *N* and *O* to engage, with certainty. When the crosshead is lifted, therefore, the mixing-valve will be lifted with it, until, at the proper point of the stroke determined by the governor, it becomes disengaged. The unhooking is accomplished by the pull-back link, *P*, in connection with a small cam, *C*, on the governor shaft, *G*. The reciprocal action between these two members is readily understood from the figure. The end of the pull-back link will, of course, oscillate with the crosshead and the hook, *H*, to which latter it is hinged; and, from the position in which it is shown in the figure, it can swing only a small angle concentrically with the governor-shaft. If given a greater angular swing the cam-rider, *R*, will ride up on the cam, and carry the link eccentrically in a direction away from the valve-stem head. Therefore, when the crosshead *X* is lifted the valve will follow its motion, until the pull-back link assumes such an angle that the cam-rider is brought far enough up on the cam to cause the hook to be pulled away from the catch-block, *O*, on the valve-stem head. When the governor rises the cam is carried in the direction of the arrow, causing the hook to be pulled out at an earlier point of the stroke; finally so early as never to lift the valve at all. When the hook is pulled, the valve drops to its seat, due to the combined action of gravity and the tension of the spring, *Q*, but, in seating, it will be cushioned by the cushion-piston, *Y*.

While the mixing-valve is open the air and gas pass from the upper and lower valve-chambers and mix in the intermediate mixing-chamber, *D*, and from there the mixture passes through the inlet port to the cylinder. At *E*, Fig. 165, is a hand-wheel for

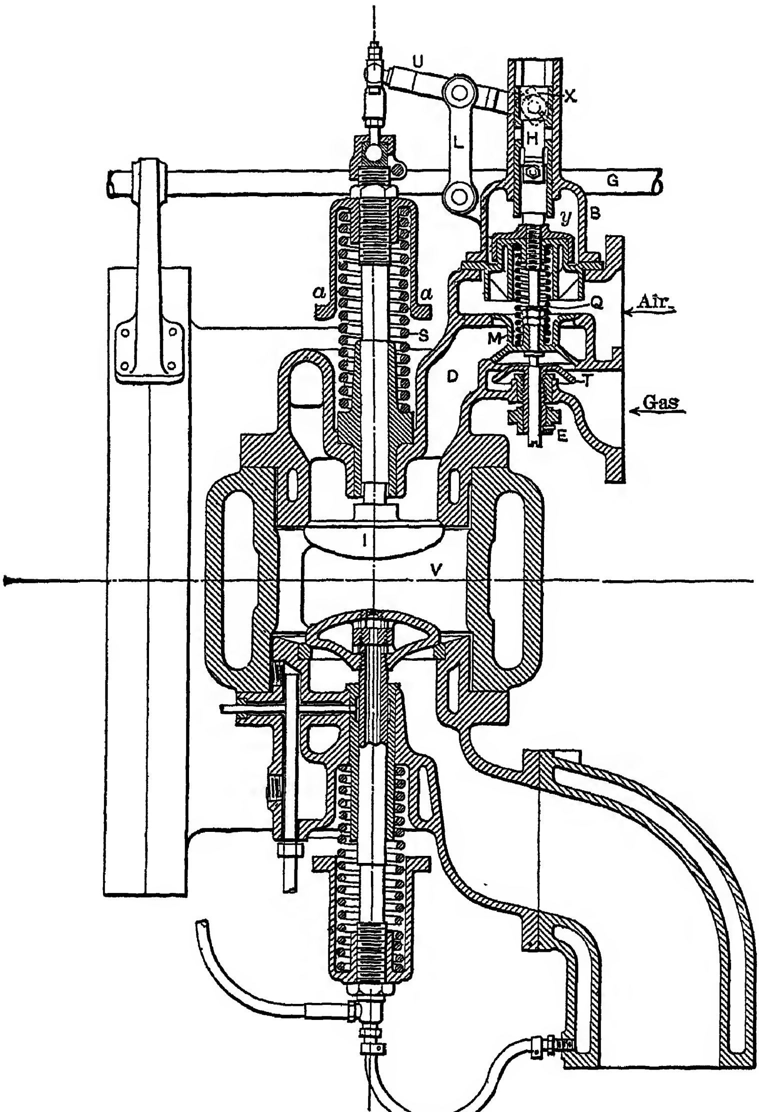


FIG. 165.—The "Snow" Standard Valve Gear. Longitudinal Section.

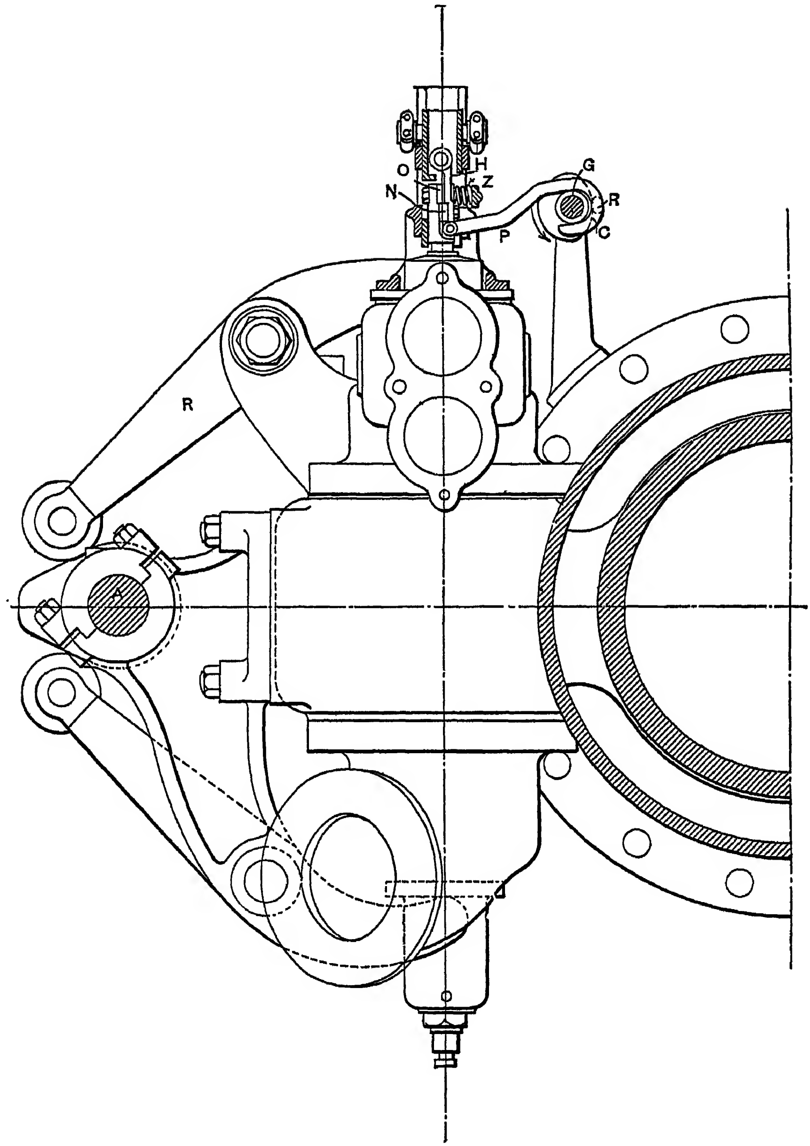


FIG. 166.—The "Snow" Standard Valve Gear. Cross-section through Cylinder.

TABLE

Reports on Gas-

Name of Installation.	Name of Experimenter.	Locality.	Year.	Cylinder. Dia.	Cylinder. Str.	Rev. per min.	Average M.E.P. Lbs. per sq. in.	Power. I. H.P.	Power. B.H.P	Mech. Eff'y.
Illuminating Gas—English Engines.										
National	Robinson	Ashton	1898	10	18	170	87	26.4	23	87
Tangye	Witz	Lille	1902	10	19	193.6	79	28.6	24.5	85
Crossley	Atkinson	Openshaw	1894	11.5	21.0	173	98	46.5	39.9	86
Crossley	Clerk	Openshaw	1894	7	15.0	200	96	14	12	86
Stockport	Bellamy	Reddish	1895	9.4	17.0	184	90	24.5	20.8	85
Illuminating Gas—French and German Engines.										
Deutz, Otto	Meyer	G. M. Fabr Deutz	1898	15	22.8	199	73	73 2	65.1	89
Koerting	Meyer	Hanover	1899	6.8	13.3	221	78	11.7	10.4	89
Niel	Witz & Moreau	Evreux	1901	13.7	18.9	213	71	53.5	46	86
Gyldner	Schroeter		1904	10	15.7	210.7	106	35.9	......	
Natural Gas.										
Westinghouse	Millar & Gladden	Pittsburg	1899	25	30	147	61	677	606	90
Westinghouse	Robertson	Lafayette	1900	13	14	265	60	113	92	79.4
Snow Pump W.	Hastings & Parker	Halsey	1901	25	48	....	......	736	595	80.7
Coke-Oven Gas.										
Borsig-Oechelhaeuser	Meyer	Silesia	1905	26.6	37.5	110.6	75	Total 878 net 765	628	Total 71 5 net 82.1
Producer Gas.										
Mond Gas Premier Eng.	Humphrey	Winnington	1900	28.⅛	30	128	81	489	368	Total 76 net 81.2
Koerting Gen. Koerting Eng.	Meyer	Hanover	1900	21.6	37.7	101	68	481	341	71
Dawson Gen. Grossley Eng.	Meyer	Zurich	1895	16.9	24	165	58	64	58	90
Westinghouse Eng. Loomis-Pettibone Gen.	Alden & Bibbins	Worcester	1907	23.5	33	150	55	600	500	83.5
Blast-Furnace Gas.										
Cockerill	Hubert	Seraing	1900	51	55	93	56	886	725	82
Deutsche Kraft A. Gesellschaft	Meyer	Differdingen	1898	17	27.5	160	63	79.5	67.6	85
Nuernberg	Riedler	Rombach	1904	33.5	43.4	105.6	70	1427	1186	83.1
Oil Engines.										
Banki	Taborsky	Buda Pest	1901	9.8	15.7	209	......	......	26.38	.....
Diesel	Ade Clark	Ghent	1903	.....	.....	154	......	203	164	80.7
Diesel	C. Eberly	Augsburg	1908	.....	.....	402.4	......	392	297	76.2
Diesel	Meyer	....	1903	15.75	23.65	158.8	95	88	69.6	79.1
Alcohol Engines.										
Deutz	Meyer	....	1903	8.35	11.8	276.9	......	16.8	......	......
Marienfelde	Meyer	....	1903	9.95	15.75	197.6	......	19.77	......	......
Banki	Schimanek	Buda Pest	1903	.....	.....	225	......	......	32.13	......

Engine Performances.

Heating Value of Gas B. T. U. per c. f. or Lbs.	Fuel Used Per Hour. per I.H.P. c. f.	Fuel Used Per Hour. per B.H.P. c. f.	Heating Value Used Per Minute. per I.H.P. B.T.U.	Heating Value Used Per Minute. per B.H.P B.T.U.	Thermal Efficiency B.T.U. per I.H.P.	Thermal Efficiency B.T.U. per B.H.P.	Engine Cycle.	Type. Single or Double Acting.	Reference.
630	14.0	16.1	148	170	28.8	25.0	4	S	Rep. Pr. Robinson.
609.7	14.5	17	147.3	172.7	28.8	24 5	4	S	Aimé Witz.
640	14.5	16.5	155	176	28	24	4	S	Eng'g. Nov. 30, 1894.
680	14.8	17	168	193	25.6	22	4	S	P. I. C. E. Vol. 124.
642	19.0	22.3	203	238	21.2	18	4	S	P. I. C. E. Vol. 124.
557.5	13.7	15.4	127.3	143	33.3	29.6	4	S	Z. V. D. I. Vol. 43.
496.5	16.0	18.0	132.4	149	32.0	28.4	4	S	Z. V. D. I. Vol. 43.
636.6	13.3	15.5	141	164.5	30.3	26	4	S	Aimé Witz.
....	......	......	......	......	42.7	......	......	......	Z. D. V. I. Vol. 48.
H. H. V. 1000	8.9	10	148.3	166.6	28.6	25.5	4	2 Cyl. D	Sibley Journal, 1900.
H. H. V. 970	10.62	13.69	171.6	221.2	24.7	19.3	4	3 Cyl. S	Bryan Donkin and Trans. A. S. M. E. 1900
H. H. V. 1175	7.37	9.13	144.3	178.8	29.4	23.7	4	4 Cyl. S	Sibley Journal, 1901.
380	17.8	25	Per T. 112 Per N 128	157	Total. 37.9 net. 33	27.5	2	1 Cyl. D	Z. V. D. I. Vol. 49.
144	52	69	125	166	33.7	25.6	4	2 Cyl. S. T.	Proc. I. M. E. 1901.
129	57.9	81.5	125	176	34	24.1	2	D	Gas L. Jour. 1900.
144	1.22 lbs.	* 1.36 lbs.	......	......	24	21.5	4	S	Z. V. D. I. Vol. 39.
106	79	95	140	168	30.1	25.2	4	2 Cyl. D	Trans. A. S. M. E. 1907.
97	83	101	134	163	31.5	26.0	4	D	Bul. de la Soc. Ind. Min. Vol. 14.
105	79	03	139	164	30	25	4	S	Z. V. D. I. Vol. 43.
88	85	102	125	150	33.9	28.2	4	2 Cyl. DT..	Gross-Gas-Maschinen, Riedler.
Benzine 18300 per lbs.	......	0.48 lbs.	......	148	......	28.5	4	S	The Engineer, 1903.
Crude oil 19300 per lbs.	0.33 lbs.	0.41 lbs.	107	131	40	32.3	....	......	Proc. I. M. E.
Kerosene 18130 per lb.	0.302 lbs.	0.43 lbs.	91.2	130	45.8	32.2	4	4 Cyl. S Cr. 180°	Z. V. D. I. Vol. 52.
Kerosene 18610 per lb.	......	......	......	......	40	31.9	4	S	Z. V. D. I. Vol. 47.
9900 per lb.	......	0.813 lbs.	......	.....	....	31.6	4	S	Z. V. D. I. Vol. 47.
9900 per lb.	......	0.786 lbs.	......	.....	.. ..	32.7	4	S	Z. V. D. I. Vol. 47.
9700 per lb.	......	0.87 lbs.	......	......	.....	30.2	4	S	Z. V. D. I. Vol. 47.

the adjustment of the throttle-valve, T, in the gas-port, by which the proportioning of the gas and air is effected.

Operating as described, this regulation effects a constant-quality mixture, but, if desirable, in order to obtain a more constant compression, the mixture may be diluted at light loads, by arranging, in the gas-supply pipe, a butterfly valve that will be controlled by the governor.

The Cockerill Engine.—Another well-known double-acting, four-cycle engine of the same general type as the ones described in the previous is the Cockerill engine, built by the Cockerill Co. of Seraing, Belgium. Similarly to the Nuernberg engine, and to German engines in general, the Cockerill construction employs a main centre-crank, which allows the shaft, as well as the main engine frame, to be built lighter than would be required for an overhung crank. The construction involves, however, difficulties in the maintaining of the shaft-journals in proper alignment, that must be taken into consideration when judging the relative advantages between the American and European practice in this respect.

In Table XXXI are recorded some performances of gas-engines of various types and on different fuels. The highest efficiencies recorded for each group are, approximately, the best that has been obtained for the particular fuel, whereas the low figures, in the majority of cases, represent efficiencies that may be expected under ordinary good conditions, and at full load.

The average M.E.P. has been computed from the power generated and the total volume, per minute, of the pressure strokes. Generally, the tests recorded have been made under approximately full-load conditions; therefore, the M.E.P. is, in most cases, the maximum. In some cases, however, the M.E.P. is very low, showing that the average load has been below the maximum, or a poor mixture has been used.

The alcohol engines recorded in the table are all of high efficiency, due to a very high compression of the charge, that has been made possible by injecting water in to the cylinder with the fuel-charge.

The compression ratios employed for the three alcohol engines recorded are:

The Deutz engine 1 to 8.9.

The Marienfelde engine 1 to 10.26.

The Banki engine 1 to 10.

Denatured alcohol was used in all three tests; in the two first the hydration was 91.2 per cent alcohol, by volume, and in the third it was 87 per cent alcohol, by volume.

CHAPTER XV

PRODUCER-GAS AND GAS-PRODUCERS

Introductory.—Producer-gas was used, as early as 1857, by Siemens as fuel in his steel-furnaces, and in connection with his regenerative furnace it was found to be a most convenient and economical means for obtaining the high temperatures which he required in his steel-making process.

Although economical as fuel, the Siemens gas is not efficient for use in the gas-engine. For the reason that, when used as fuel, not only the potential heating-value of the gas, but also its sensible heat, becomes available heating-value; whereas, as the gas can be used in the gas-engine only after having been cooled, a great percentage of its total heating-value is thrown away in the cooling-process.

The year 1881 can be counted the beginning of the era of producer-gas power. Mr. Dawson exhibited that year his first successful power-gas producer, in connection with a small Otto gas-engine, and the success of the exhibit was such that producer-gas became, soon afterward, to be considered a most efficient competitor with steam as a means for generating power from available fuels.

Since then, improvements have been made in the gas-producer as well as in the gas-engine, until, to-day, producer-gas is extensively used with success, in many instances on a large scale in preference to steam.

The principal difference between the way in which Siemens generated his fuel-gas and the way in which power-gas is obtained is, that in the modern gas-producer steam is introduced into the furnace, for the purpose of cooling the fuel-bed to some extent, and thus prevent an excessive loss of heat in sensible heat of the gases, that will be wasted in the subsequent cooling-process.

The Gas-Producer.—The simple suction gas-producer, shown

in Fig. 167, as well as, in a general way, the pressure producer, may be described as an air-tight steel-plate cylinder containing at the bottom an ash-pit, a grate for supporting the fuel, sometimes a shaking-grate, and, above, a substantial firebrick-lined furnace. Above the furnace there is a fuel-hopper, and a charging bell seals the opening through which fuel is charged into the producer. There are also poke-openings provided on top of the producer-furnace, through which poke-bars may be inserted for the purpose of poking down or breaking up the fuel-bed, which may be held up or made solid by the caking or clinkering of the fuel. At a suitable height above the grate, there is a gas-outlet taken for conveying the generated gases from the producer in to the necessary cooling and cleaning apparatus.

When in normal operation, there is on the grate of the furnace a thin bed of ashes and, above, a deep incandescent fuel-bed on top of which a layer of fuel that has not as yet attained high incandescence should preferably be in evidence.

The only access for air for maintaining combustion in the producer is generally through the ash-pit, and the draft through the fuel-bed must be regulated to that necessary for generating the required amount of gas of a proper quality. The draft is maintained either by the suction of the gas-engine piston, by a pressure-blower, or by an exhauster of some kind.

The Process.—The result of this arrangement of blowing air in to, or drawing a current of air through, a deep bed of incandescent fuel will be as follows:

Near the grate-level there is formed, through ordinary combustion, carbon dioxide; a gas often referred to as CO_2-gas, and which, as its name implies, is composed in such a manner that each molecule, or element of the gas, consists of one atom of carbon and two atoms of oxygen.

On being drawn through a deep, porous bed of highly heated carbon, this gas absorbs readily more carbon, so that new molecules, consisting each of one atom of carbon and one atom of oxygen, are formed. This new gas, carbon monoxide or CO-gas, which is always the result of incomplete combustion of carbon, is the main constituent of producer-gas.

By incomplete combustion of one pound of carbon there is formed $2\frac{1}{3}$ pounds of carbon monoxide gas, and the heat generated is about 4,380 heat units. The carbon monoxide gas formed from each pound of carbon, at the primary combustion in the producer, requires for its combustion the same amount of oxygen as the original pound of carbon consumed at its gasification to CO, or $1\frac{1}{3}$ pound, and the heat generated by the combustion is about 10,200 heat units. It will, thus, be seen that the main combustible of producer-gas contains a potential heating-value of about two-thirds of the heating-value of the fuel, if the latter, as has been assumed, consists of pure carbon only.

The carbon monoxide is, however, not all the heating-value obtained in the gas at the primary gasification, because the hydrocarbons which most fuels contain, and which are distilled off into the gas, enriches it, and, further, the sensible heat of the gas adds materially to its total heating-value.

At the primary combustion there is liberated, as stated, about 4,380 heat units for each pound of carbon gasified to CO-gas. This heat will, of course, tend to heat up the fuel-bed and the producer, until the radiation from the apparatus, together with the sensible heat that is carried off by the gases, will just balance the heat liberated.

The Siemens producer for fuel-gas was operated according to the preceding outline, and it generated a gas low in potential heating-value but of high temperature. That is, the gas carried off, when leaving the producer, a considerable amount of heat as sensible heat.

In the modern producer the object is to generate combustible gas of as high heating-value as possible, when cold, but any sensible heat is not required for the process, excepting to the amount necessary for maintaining the temperature of the furnace at such degree that the formation of CO-gas takes place readily. For the formation of combustible gas the producer must be supplied with the amount of air, only, that is required for the gasification of carbon to carbon monoxide gas. But only part of the heat generated during this gasification, necessarily only about

1,900* heat-units, will be carried away by the gases, or dissipated through radiation, when the temperature of the furnace is such as is normally required for efficient gasification. There will, therefore, be a surplus of heat, about 2,500 heat-units per pound of coal consumed, that must be abstracted from the fuel-bed, in order not to overheat the same as the process of gasification proceeds. This surplus heat can be utilized in a most efficient and desirable manner, simply by the introduction of steam in to the furnace.

Steam, being subject to decomposition into its constituent elements, oxygen and hydrogen, when in contact with incandescent carbon of sufficiently high temperature, is a most useful agent in a gas-producer. It serves, primarily, three good purposes; in that it absorbs the surplus heat from the furnace at its decomposition, in that it enriches the gas with its hydrogen, and in that it furnishes oxygen for the gasification of carbon.

The oxygen which is furnished to the gas-generator by the introduction of air carries with it nitrogen in the proportion of 3.33 pounds of nitrogen for each pound of oxygen, and this nitrogen, which dilutes the gas, will be of no value as far as the heating-value of the gas is concerned. It is, therefore, evident that the more oxygen that can be obtained for the combustion of carbon by decomposition of steam the richer in heating-value the generated gas will be.

There are, however, only about 2,500 heat-units available for the decomposition of steam and for the formation of hydrogen, and, as 327 heat-units are required for the formation of each cubic foot of hydrogen, there is a limit of 8 cubic feet of hydrogen that can be obtained per pound of carbon by utilizing the surplus heat only for its formation.

As from each pound of carbon there will be generated 2⅓ pounds, or 32 cubic feet of carbon monoxide, the volume of hydrogen will, therefore, be about one-fourth of that of the carbon monoxide, when all the fuel is burned to this gas.

*This figure is approximate only; its value having been estimated with reference to the average conditions obtaining at an efficient gas-making process as carried out at present. See page 423.

Generally we find, however, producer-gas in which the volume of hydrogen is more than one-fourth of that of the carbon monoxide, and this may be taken as evidence that some of the fuel has been subjected to complete combustion to carbon dioxide-gas. When the volume of the carbon dioxide in producer-gas is not over five per cent, then there has been no perceptible loss incurred by having burned the carbon completely to carbcn dioxide, because the additional heat generated thereby has been utilized in forming a corresponding additional percentage of hydrogen, which enriches the gas.

What we call producer-gas consists, as will appear from what has been stated, principally of five separate elementary products, each of a very distinct nature.

These are:

Carbon monoxide, which is obtained at the primary gasification of carbon;

Hydrogen, which has been formed by reclaiming the heat liberated at the primary gasification by decomposition of steam;

Hydrocarbons, which have been distilled off from the fuel;

Nitrogen, obtained from the air consumed, and

Carbon dioxide gas formed at the complete combustion of some of the fuel.

Of these elementary products, the three first are combustible gases, and they compose somewhat more than 40 per cent of the total volume of the producer-gas.

Nitrogen and carbon dioxide are inert gases and compose nearly 60 per cent of the total volume of the generated gas.

Producer-gas has in some rare cases been found to contain, when hot, as much as 93 per cent of the heating-value of the fuel from which it has been formed, but, when cooled and ready for use in the gas-engine, it carries rarely more than 85 per cent of the heating-value of the fuel.

The sensible heat of the gas when it leaves the producer is, in part, generally utilized for heating the fresh fuel charge, for pre-heating the air and for vaporizing the steam that is supplied to the gas generator.

Heat-Transfer of a Theoretical Gas-Making Process.—The

waste of fuel in the ashes, and the waste of heat through radiation are the only direct losses of heating-value in the gasification of fuel. In the process itself there is no loss or gain, only a transfer of heating-value, from the carbon, in to the gases generated. Theoretically, the gasification may be made, with equal economy, either into carbon monoxide and hydrogen or into carbon dioxide and hydrogen, and that this is so will readily be seen by the following representations of the heat-transfers that must be due to the reactions taking place in the two cases.

It is assumed, in the first case, that 12 pounds of carbon are gasified by the oxygen derived through dissociation of 18 pounds of water into 2 pounds of hydrogen and 28 pounds of *C O*-gas.

The total potential heating-value of 12 pounds of carbon is $12 \times 14{,}600 = 175{,}200$ B.T.U., of which:
there are évolved, at the combustion of 12 pounds of carbon to *C O*-gas, $12 \times 4{,}380 = 52{,}560$ B.T.U.;
the balance, 122,640 B.T.U., is transferred to the *C O*-gas.

The dissociation of 18 pounds of water requires $18 \times 6900 =$ 124,200 B.T.U., and of this amount 52,560 B.T.U. become available at the combustion of the carbon, the rest, 71,640 B.T.U., must be supplied from outside source. The total amount, 124,200 B.T.U., absorbed at the dissociation of the water will be found as potential heating-value in the hydrogen formed.

The figures above are based on the high calorific value of hydrogen, and include therefore the heat required for the vaporization of the water into steam.

When carbon is gasified into *C O*-gas and hydrogen the following theoretical heat-transfer will take place:

12 lbs. C +	B.T.U.	18 lbs. H_2O =	B.T.U.	2 lbs. H_2 +	B.T.U.	28 lbs. CO	B.T.U.
Potential value.	175,200	Absorb at dissociation:					
Evolved at comb. to CO	52,560	From outside source.....	71,640				
Transferred to CO-gas	122,640	From comb. of C.......	52,560	Potential value.	124,200	Potential value.	122,640
		Transferred to H_2	124,200				

Hence $\frac{52,560}{124,200}$ (= 0.42) of 18 pounds of water is dissociated by heat evolved at the combustion of 12 pounds of carbon to CO-gas, and 58 per cent must be decomposed by outside heat-supply, or the heat evolved from each pound of carbon will dissociate 0.63 pound of water.

It is evident, therefore, that, at a theoretical gasification, 42 per cent of the carbon will be gasified by oxygen liberated from water due to the heat evolved from the carbon itself, and 58 per cent must be gasified by oxygen derived from the atmosphere.

If, secondly, the carbon is gasified to CO_2-gas by oxygen derived from water, then 12 pounds of carbon will require the dissociation of 36 pounds of water to form 4 pounds of hydrogen and 44 pounds of CO_2-gas.

In the gasification of carbon into carbon dioxide and hydrogen, the theoretical heat-transfer will be as follows:

12 lbs. C +	B.T.U.	36 lbs. $2H_2O$ =	B.T.U.	4 lbs. $2H_2$ +	B.T.U.	44 lbs. CO_2	B.T.U.
Potential value....	175,200	Absorb at dissociation:					
Evolved at comb. to CO_2	175,200	From outside source.....	73,200				
Transferred to CO_2-gas..	0	From comb. of C.	175,200				
		Transferred to H_2.	248,400	Potential value.	248,400	Potential value.	0

Hence $\frac{175,200}{248,400}$ (= 0.70) of 36 pounds of water is dissociated by heat evolved at the combustion of carbon to CO_2-gas and 30 per cent must be decomposed by outside heat-supply; or by the heat evolved from each pound of carbon there will be dissociated 2.1 pound of water.

Thus, 70 per cent of the carbon will be gasified by oxygen liberated from water due to the heat evolved from the carbon itself, and 30 per cent must be gasified by oxygen derived from the atmosphere.

Composition of Gas Resulting from Gasification without Heat

Loss.—A theoretical gasification of one pound of C to H, $C\,O$ and N will result in the following composition:

	Pounds.	Per Cent. Weight.	Cubic Feet.	Per Cent. Volume.
$C\,O$	2.333	0.47	31.64	0.40
H	0.07	0.015	13.30	0.17
N	2.57	0.515	34.85	0.43
	4.97	1.00	79.79	1.00

Heating-value per cubic foot 184 B.T.U.

The theoretical composition of a gas resulting from the gasification of one pound of C to $C\,O_2$, H and N will be the following:

	Pounds.	Per Cent. Weight.	Cubic Feet.	Per Cent. Volume.
$C\,O_2$	3.67	0.56	31.64	0.28
H	0.234	0.036	44.27	0.40
N	2.66	0.404	36.07	0.32
	6.56	1.000	111.98	1.00

Heating-value per cubic foot 130 B.T.U.

An examination of the weights of the elementary gases obtained in each case will show, that the heating-value of the total resulting composition is in both cases the same; the heating-value of 2.333 pound of $C\,O$ being the same as that of 0.164 pound of H.

The weight of the resulting gases is, however, in the latter case considerably more than in the former.

During the practical process of generating producer-gas there is always some loss, due to heat-radiation and due to cooling of the gas, and the greater the weight of the final product is the greater will be the heat-loss incurred. It is evident, therefore, that the process of generating H and $C\,O_2$-gas will involve a greater loss than the process of generating H and $C\,O$-gas.

In the following tables are given the composition and heating-value of producer-gas generated from carbon, with varying percentages of the fuel burned to $C\,O_2$-gas. The assumption has been made that in producing $C\,O$-gas an efficiency of 85 per

cent is obtained, and that in producing CO_2-gas and H the efficiency will be 70 per cent.

Composition of Producer-Gas Containing Varying Percentages of CO_2.—Increasing percentages of the fuel are assumed to have been burned to CO_2-gas, with correspondingly diminished efficiency.

Per Cent C Gasified to		Products Derived from 1 lb. C, in Pounds.				Total Weight of Products from 1 lb. C, in lbs.	Heating Value per pound of Gas.
CO.	CO_2.	CO.	H.	N.	CO_2.		
100	0	2.333	0.035	3.5	0	5.868	2120
95	5	2.216	0.042	3.55	0.183	5.991	2053
90	10	2.099	0.048	3.6	0.367	6.114	1990
85	15	1.982	0.055	3.65	0.55	6.237	1935
80	20	1.865	0 061	3.7	0.733	6.359	1882
75	25	1.748	0.068	3.75	0.917	6.483	1830

Per Cent C Gasified to		Products. Percentage Volume.				Volume of Products from 1 lb. C, in cub. ft.	Heating Value per cub. ft. B.T.U.
CO.	CO_2.	CO.	H.	N.	CO_2.		
100	0	37.0	7.7	55.3	0	85.7	145
95	5	43.3	9.0	54.9	1.8	87.7	140
90	10	31.8	10.2	54.5	3.5	89.6	136
85	15	29.4	11.3	54.1	5.2	91.5	132
80	20	27.1	12.4	53.7	6.8	93.4	128
75	25	24.9	13.4	53.4	8.3	95.3	125

Quantity of Steam to be Supplied.—Eighty-five per cent is on an average the best return in heating-value obtained in a cold gas generated from an anthracite fuel of, say, 14,000 B.T.U. The heat-loss per pound of fuel is, thus, 2,100 B.T.U.

The principal wastes are: (1) in the sensible heat carried off by the gases; (2) in loss of fuel in the ashes, and (3) in heat dissipated through radiation.

An approximation of the amount of the first item is readily obtained, as it is known that, normally, 5½ pounds of gases are

produced per pound of carbon, and that they leave the producer, normally, at about 850° F. Their specific heat is, on an average, 0.25. The air and vapor being supplied at a temperature of, at least, 150 degrees, the range of temperature through which the gases are heated in the furnace is about 700 degrees.

The heat-loss, therefore, due to item (1), is 962 B.T.U. per pound of fuel. The waste of fuel should not exceed 1½ per cent, or 210 heat-units per pound of fuel. The total loss due to items (1) and (2), thus, 1,172 B.T.U. If this heat-loss be subtracted from the total loss 2,100 B.T.U. there remains 928 B.T.U., which is the loss due to radiation. Hence, it may be said that approximately 1,900 B.T.U. are wasted in the cooling of the gases and through radiation. The approximate value of this item of heat-loss is of interest, as it has bearing on the quantity of steam that should, normally, be supplied to the furnace.

Deducting 1,900 B.T.U. from the heat evolved at the gasification of the fuel to CO, 4,380 B.T.U., the remaining 2,480 B.T.U. becomes available for decomposition of steam.

It has been found, however, that for the reduction of all the carbon dioxide formed in the lower part of the furnace it is necessary that the temperature of the fuel-bed be very high, and the current through it slow. A high furnace temperature will, on the other hand, result in increased loss due to the higher temperature at which the gases are carried off and in increased radiation losses. Besides, difficulties will result from clinkering of the refuse in the fuel.

Practice has shown that an average temperature of the fuel-bed of 1,800 to 2,000 F°*. will give best results, but, at such low furnace temperature, there will remain in the gas an appreciable amount of CO_2-gas, unreduced.

In an average, favorable case seven per cent of the carbon will be burned to CO_2-gas and hence an additional heating-value of 715 B.T.U. per pound of fuel will be available for decomposition of steam. The total available heating-value, thus, 3,195 B.T.U.

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* Compare: "Producer Gas," by J. Emerson Dawson, pp. 6 and 45.

The weight of the steam to be supplied is, therefore, $\frac{3,195}{6,900}$ ($= 0.463$) pounds, per pound of fuel. At the decomposition of this steam there is formed $\frac{8}{9} \times 0.463$ ($= 0.412$) pound oxygen, which will effect a gasification of $\frac{0.412}{1.33}$ ($= 0.31$) pound of carbon.

Accordingly, there will be required normally not more than 0.46 pound of steam per pound of carbon, and the oxygen derived at its decomposition will gasify, on an average, 0.31 pound of carbon.

The specific heat of steam being high, a material loss will be incurred by supplying an excess amount of steam, above what can be decomposed by the fuel. The excess will only, in passing through the fuel-bed, absorb heat as superheat, which will be wasted at the subsequent cooling of the gases. Besides, it will cool the lower part of the fire below the required temperature for efficient reduction of the CO_2-gas, and after the fire has been driven to the top of the fuel-bed due to excessive steam-supply, there will generally be experienced some little difficulty to get the fuel-bed in condition again for producing a proper gas.

Exhaust-Gas from the Engine used in the Producer Instead of Steam.—Instead of using steam as a means for reclaiming the heat generated at the primary combustion in the producer, the exhaust gases from the gas-engine have been successfully employed for the purpose. The process consists in the reduction of the CO_2-gas, returned to the producer from the engine-exhaust, into CO-gas. The gas generated by this method, consisting principally of CO-gas and nitrogen, will stand a high compression, but, being very lean in heating-value, it will require somewhat larger cylinder capacity than ordinary producer-gas. On this account, it is doubtful if the increased cost of the installation does not outweigh the gain due to any increased efficiency that may be derived through the employment of a high compression.

The saving due to the utilization of the sensible heat of the exhaust gases cannot be material, particularly as it is seldom

convenient to locate the engine close enough to the producer to fully take advantage of this item; moreover, the gasification process requires the return to the furnace of only a part of the total volume of the exhaust, and in it is included a considerable amount of nitrogen that becomes reheated in the furnace, and generally leaves the producer at a higher temperature than that at which the exhaust gases are admitted.

It can readily be ascertained that, theoretically, the gas generated by the method of by-passing the exhaust from the engine in to the furnace can never be richer in heating-value than gas obtained through gasification with air only, according to Siemens process. The heat of the primary gasification can be reclaimed, but the heating-value, per cubic foot of gas, will not be enriched thereby. A demonstration, as the following, will show this.

For each cubic foot of CO-gas generated by the combustion of carbon, figured at 62° F., there will be evolved 140 B.T.U. Assuming all this heat to be utilized for the decomposition of CO_2 of the exhaust, then, as there is required 324 B.T.U. for the decomposition of one cubic foot of CO_2 into one cubic foot of CO and ½ cubic foot of O, there will be obtained

$$\frac{140}{324}(1\ c.f.\,CO + \tfrac{1}{2}\ c.f.\,O) = 0.43\ c.f.\ CO + 0.215\ c.f.\ O.$$

Each cubic foot of CO-gas formed by combustion of C with air will carry 1.88 cubic foot of nitrogen, whereas each cubic foot of CO reduced from CO_2-gas will carry twice this amount, or 3.76 cubic feet. This is evident because the CO_2 in the exhaust has consumed twice as much air as that required for the combustion of carbon to CO-gas. The process that will take place at the gas-production will be the following.

At the production of 1 cubic foot of CO-gas from carbon there will be generated:

0.57 *c.f.* of CO by air, carrying $0.57 \times 1.88\ N$ =	1.07 N.
0.43 *c.f.* of CO by O, carrying	*no* N.
0.43 *c.f.* of CO reduced from CO_2, carrying $0.43 \times 2 \times 1.88\ N$ =	1.62 N.
1.43 *c.f.* of CO in total, carrying	2.69 N.

Hence, each cubic foot of CO will carry 1.88 cubic foot of N, or, in other words, the heating-value per cubic foot of the products will be at par with that of air-gas.

The above computation may appear complicated, but it can be simplified if the following facts are recognized:

That 1 cubic foot CO-gas obtained by the gasification of C with air carries 1.88 cubic foot N.
That 1 cubic foot CO_2-gas of the exhaust will carry at least 2×1.88 cubic foot N;
That ½ cubic foot O will give with C 1 cubic foot CO.

Assuming, then, that, through combustion of carbon with air and with O, there has been evolved, in the furnace, heat enough for the decomposition of just one cubic foot of CO_2, thus,

1 cubic foot of CO_2 becomes decomposed to	
1 cubic foot of CO, carrying	2×1.88 N,
and ½ cubic foot of O, giving with C	
1 cubic foot of CO, carrying	*no* N.

Hence, the advantage gained in obtaining free oxygen through dissociation of CO_2-gas, is exactly balanced by the increased amount of nitrogen which the CO_2-gas carries in to the furnace.

It will be evident that for reclaiming all the heat of the primary combustion there will be required $\left(\frac{43}{143} = \right)$ one-third of the total volume of the exhaust from the engine.

Ninety cubic feet of Siemens air-gas generated from one pound of carbon contains 10,200 B.T.U., or the gas is of 113 B.T.U. per cubic foot.

There is added in ordinary producer-gas, theoretically, 8 cubic feet of hydrogen of a heating-value of, in all, 2,500 B.T.U., making the gas of a heating-value of 130 B.T.U. per cubic foot, or 15 per cent richer than air-gas. Practically, due to partial combustion of the fuel to CO_2, by which an additional volume

of hydrogen is formed, the ordinary producer-gas becomes richer still.

On the other hand, when the difficulty of regulating the volume of the exhaust returned to the furnace to the correct proportions is considered, it may be expected that, practically, the gas produced when the engine exhaust is returned to the generator would become even of a comparatively poorer quality than the theoretical computation tends to show. But the fact is that the gas, actually, will be enriched by an appreciable amount of hydrogen derived from the moisture in the fuel and in the air charged to the furnace, as well as from the moisture in the air charged to the engine, which will appear in the exhaust gases.

The best efficiency so far reported to have been obtained by this gas is about at par with that obtained by ordinary producer-gas.*

The Composition and Heating-Value of Producer-Gas.—Theoretical Analysis of Anthracite Gas.—The normal composition of producer-gas may be determined, approximately, from the analysis of the fuel from which it is made.

With respect to the hydrocarbon contents, the composition of anthracite gas will be approximated by assuming that all the hydrocarbons are distilled off from the fuel, and that they will be found in the gas in the form of marsh-gas.

Assume the analysis of an anthracite coal to be:

Fixed carbon	84 per cent
Volatile hydrocarbons	4 per cent
Ash and moisture	12 per cent

It is safe to assume that 25 per cent of the fixed carbon will be gasified to CO-gas by oxygen derived from steam, 65 per cent gasified to CO-gas by oxygen derived from the atmosphere, and that 10 per cent is burned to CO_2-gas.

* Compare the efficiency reported by Mr. G. M. S. Tait, page 795, Transact. Am. Soc. Mech. Eng., Vol. XXX, with producer-gas reports, Table XXXI.

	Pounds of Gas.	Per Cent. Weight.	Cub. Feet Gas at 62° F.	Per Cent. Volume.
Hence, of 100 *pounds of fuel:*				
21 lbs. C gasified to CO by steam require 28 lbs. O, giving CO	49.0			
54.6 lbs. C gasified to CO with air require 72.8 lbs. O, giving CO	127.4			
TOTAL CO.......	176.4	33.22	2392.0	31.04
8.4 lbs. C burned to CO_2 require 22.4 lbs. O, giving CO_2.....	30.8	5.80	265.8	3.45
4. lbs. hydrocarbons distilled give CH_4....................	4.0	0.75	94.9	1.23
88 lbs. combustible.				
28 lbs. O dissociated from steam liberate H	3.5	0.66	665.0	8.63
95 lbs. O derived from the atmosphere carry N.............	316.3	59.57	4289.0	55.65
	531.0	100	7706.7	100

The heat-energy in the gas will be

in 176.4 lbs. CO at 4,380 B.T.U. per pound	772,632	B.T.U.
in 4 lbs. CH_4 at 21,900 B.T.U. per pound	87,600	B.T.U.
in 3.5 lbs. H at 62,100 B.T.U. per pound	217,350	B.T.U.
Total	1,077,582	B.T.U.

Heat-energy per pound gas 2,030 B.T.U.
Heat-energy per cubic foot at 62° F. 140 B.T.U.
Heat-energy in the gas from one pound of coal 10,775 B.T.U.
Heat-energy in one pound of coal 13,140 B.T.U.
The efficiency of conversion into gas 0.82

The following may be considered as an average composition of anthracite suction gas:

	Per Cent, Weight.	Per Cent, Volume.
CO	27.6	26.0
H	0.7	9.0
CH_4	1.2	2.0
N	60.5	57.0
CO_2	10.0	6.0
	100.0	100.0

Heating-value per cub. ft. at 62° F., and 1 atmosphere 134 B.T.U.
Volume per pound of gas at 62° F., and 1 atmosphere 14.5 cub. ft.

Theoretical Analysis of Bituminous Gas.—With regard to bituminous coals containing a high percentage of volatile hydrocarbons of various compositions, there exists an uncertainty as to what amount of the latter will become oxidized during the combustion, to be formed into fixed gases. Further, the gas generated from bituminous fuels is, due to condensible hydrocarbons, of a materially different composition before being cooled from what it will be after the cooling and cleaning process; there remaining vaporized in the hot gas the heavy hydrocarbons which condense at cooling.

The composition and heating-value of bituminous gas, when cold, does not in practice generally agree very well with any theoretical estimate based on the analysis of the fuel from which it has been produced. The percentage of hydrocarbons distilled over, and present in the gas, is a too important item that cannot be predicted, excepting, only approximately, by reference to some known analysis of a gas generated under conditions similar to those of the case at hand; and, further, the amount of hydrocarbons that becomes decomposed in the producer, into hydrogen and CO-gas, will depend very much on the temperature at which the gasification is made.

The reactions by which the hydro-carbons are broken up are principally the following:

$$\begin{aligned} CH_4 + H_2O &= CO + 6H \\ CH_4 + CO_2 &= CO + 4H \\ C_2H_4 + 2H_2O &= 2CO + 8H \\ C_2H_4 + 2CO_2 &= 4CO + 4H. \end{aligned}$$

In the case of fuel-gas used hot from the generator, the heating-value, including the sensible heat of the gas, approximates often 90 per cent of the heating-value of the fuel, and the potential heating-value due to the hydrocarbons can in such a gas also be assumed to be 90 per cent of that of the hydrocarbons in the fuel. The heating-value of the hydrocarbons in bituminous coal

being generally from 17,000 to 20,000 B.T.U. per pound, we obtain, thus, in the hot gas, from 15,300 to 18,000 B.T.U. per pound of hydrocarbons in the coal used.

At the cooling and cleaning of the gas there disappears on an average from 50 to 70 per cent of the hydrocarbons, in the form of tar, and the loss in heating-value in the tar, with coals varying in volatile hydrocarbons from 20 to 40 per cent, will be from 10 to 24 per cent of the heating-value of the coal.

As a standard with which to compare the results actually obtained in practice, the ideal composition of the gas, estimated on a theoretical basis, becomes often of interest.

Assume the analysis of a sample of bituminous coal to be:

Fixed carbon	52 per cent
Volatile hydrocarbons	36 per cent
Moisture and ash	12 per cent

In a favorable case 25 per cent of the fixed carbon will be gasified into *C O*-gas by oxygen derived from steam, 65 per cent to *C O* by oxygen derived from the atmosphere and 10 per cent burned to CO_2-gas.

	Pounds of Gas.	Per Cent Weight.	Cubic Feet of Gas.	Per Cent, Volume.
Accordingly of 100 *pounds of fuel:*				
13 lbs. C gasified to C O with steam require 17.33 lbs. O, giving C O	30.33			
33.8 lbs. C gasified to C O with air require 45.06 lbs. O, giving C O	78.86			
TOTAL C O.....	109.19	30.1	1,480.6	26.6
5.2 lbs. C burned to CO_2 require 13.87 lbs. O, giving CO_2....	19.07	5.2	164.6	3.0
36.0 lbs. hydrocarbons distilled, giving C_2H_4, CH_4 and H.......	36.00	10.0	854.2	15.3
88.0 lbs. combustible.				
17.33 lbs. O dissociated from steam liberate H	2.17	0.6	412.3	7.4
58.93 lbs. O derived from the atmosphere carry N.............	196.24	54.1	2,661.0	47.7
	362.67	100.0	5,572.7	100.0

The potential heating-value in the hot gas:

In 109.19 lbs. CO, at 4380 B.T.U. per pound......	478,252
In 36.00 lbs. C_2H_4, CH_4 and H, obtained from hydrocarbons, at 18,000 B.T.U. per pound ...	648,000
In 2.17 lbs. H at 62,100 B.T.U. per pound	134,757
	1,261,009

Heat energy in the gas from one pound of coal..12,610 B.T.U.
Heat energy in one pound of coal, hydrocarbons being figured at 20,000 B.T.U............ 14,800 B.T.U.

Efficiency of the conversion into gas, accordingly, 85.3 per cent.
Heating-value per cubic foot of gas, at 62° F.,....233 B.T.U.

If the average temperature at which the gases leave the producer be 1,000° F., and their specific heat 0.25, the sensible heat in 3.63 pounds of gas becomes 907 heat-units, which added to the potential heating-value of the gas brings the efficiency of the gasification into hot fuel-gas to 91 per cent.

In the cooling- and washing-process the gas would probably lose about 50 per cent of the hydrocarbons. The heat energy in the cold and cleaned gas becomes in that case:

In 109.19 lbs. CO at 4,380 B.T.U. per pound.......	478,252
In 18.00 lbs. C_2H_4, CH_4 and H obtained from hydrocarbons, at 18,000 B.T.U. per pound.........	324,000
In 2.17 lbs. H at 62,100 B.T.U. per pound.......	134,757
	937,009

The efficiency of conversion into clean gas, thus, 63 per cent.
Heating-value per cubic foot of gas, at 62° F. = 168 B.T.U.

A theoretical analysis of bituminous gas, like the above, does not always agree very closely with practice, and it may be misleading if the true percentage of the fuel-value obtained from the hydrocarbons is not known from actual gas-analyses.

Hydrocarbon Loss in Tar.—The loss in heating-value in the tar can be determined through an analysis such as the following:

From an Indiana coal containing:

Fixed carbon 52 per cent

Volatile hydrocarbons 31 per cent

there was obtained a gas as per the volumetric analysis in the first two columns of the table below.

The resulting elementary constituents are as per the last five columns of the table:

Constituents.	Per Cent. Volume.	Molecular Weight.	Ratio of Weights.	Per Cent. Weight.	Elements.				
					N.	C.	O.	H.	C H₄.
N	51	28	1428	56.9	56.9				
CO	21.5	28	602	24.0	....	10.3	13.7	1.2	......
H	15	2	30	1.2	......	......	......	......	......
CO_2	9	44	396	15.8	......	..4.3	11.5	......	2.1
CH_4	3.5	16	56	2.1	......	......	......	......	......
			2512	100.0	56.9	14.6	25.2	1.2	2.1

The volatile hydrocarbons may, without great error, be assumed to be of the average composition of methane, $C\,H_4$.

Per 100 pound of gas:

Oxygen obtained from the atmosphere is $\frac{56.9}{3.33} = 17$ pounds.

Oxygen obtained from steam is $25.2 - 17 = 8.2$ pounds.

Steam used $= \frac{9}{8} \times 8.2 = 9.22$ pounds.

Hydrogen liberated from steam $= \frac{1}{8} \times 8.2 = 1.02$ pounds.

Hydrogen derived from the hydrocarbons $=$
$1.2 - 1.02 = 0.18$ pounds.

Hydrocarbons decomposed to form hydrogen
and carbon $= 4 \times 0.18 = 0.72$ pounds.

Carbon liberated from the hydrocarbons $=$
$3 \times 0.18 = 0.54$ pounds.

Carbon derived from fixed carbon in the fuel $=$
$14.6 - 0.54 = 14.06$ pounds.

From 100 pounds of fuel, containing 52 pounds of fixed C, there is obtained $\frac{52}{14.06} = 370$ pounds of gas.

The density of the gas is $\frac{25.12}{2} \times 0.0053 = 0.0666$ at 62° F. and 1 atmosphere.

Hence, 57 cubic feet of gas is obtained per pound of fuel.

In 100 pounds of gas there is accounted for $2.1 + 0.72$ (= 2.82) pound CH_4, thus, per pound of fuel there is accounted for $2.82 \times 3.70 = 10.43$ pound CH_4, and there is lost $31 - 10.43$ (= 20.47) pound of hydrocarbons in tar.

Hence the tar is 66 per cent of the hydrocarbon values.

Production of Water-Gas.—It is possible, in any gas-producer, to generate, intermittently, gas of a considerably higher heating-value than that which a continuous process will give. The gas production is manipulated as follows: The fuel-bed is blown up with pure air to a high temperature, while the lean gases generated during the blowing-up process are wasted to the atmosphere. If then, when the fire becomes of a sufficiently high temperature, steam, or, better, superheated steam, is blown through the fuel-bed it will be possible, for a short time, to gasify the steam and carbon without any air, resulting in a gas composed of practically only hydrogen and carbon monoxide. As soon as the fire becomes too cool for decomposing the steam the gasmaking process is discontinued, and the fire blown up again.

By proceeding in this manner, wasting the gases generated during the blowing-up process and collecting the gases formed at the decomposition of steam, the resulting product will be of a heating-value approaching that of the carbon monoxide or hydrogen, or approximately 324 B.T.U. per cubic foot. This gas is commonly referred to as water-gas. The sensible heat of the product generated during the blowing-up process, consisting mainly of carbon dioxide and nitrogen, does not actually need to be wasted, as it may be utilized in regenerators for preheating the air and super-heating the steam used for the process.

The composition of water-gas is, as may be seen from the reaction-formula, page 82, of the proportion of one pound of hydrogen in 14 pounds of carbon monoxide, or equal volumes of hydrogen and carbon monoxide.

When being used as illuminating gas, this gas is enriched by addition of hydrocarbons and illuminants, until a heating-value of about 550 to 600 B.T.U. is obtained. The gas is also used for industrial and power purposes. Illuminating gas obtained

in this manner is commonly referred to as carbureted water-gas or manufactured illuminating gas.

The original Siemens producer-gas, which is obtained without the use of steam in the producer, is occasionally designated by the name air-gas, and common producer-gas, being of a quality intermediate between air-gas and water-gas, is sometimes referred to as semi-water gas.

Gas-Producers.—Gas-producers are commonly classed with reference to the nature of the fuel with which they are intended to deal; as anthracite producers, bituminous producers, and lignite and peat producers.

The anthracite producers, which have to deal only with practically tarless and non-caking fuels, such as anthracite coal, coke, or charcoal, are generally arranged as suction producers. That is, the suction of the gas-engine piston, in connection with which the producer operates, is the only means whereby the draft through the generator is induced. This arrangement insures a very simple, compact, and easily handled installation.

The operation of the bituminous producers has, due to the nature of the fuel, more difficulties connected with it than the simple anthracite producer offers. These difficulties are mainly due to the caking of the fuel in the generator, which prevents the free flow of the gas-current through the fuel-bed, and also due to the formation of large quantities of tar in the process of cooling the gas.

The caking of the fuel is generally counteracted, as much as possible, by frequent poking operations, which have for object to keep the fuel-bed in a sufficiently porous state to allow the gases to pass, and escape freely.

To rid the gas of tar and dust there are employed a variety of static and mechanical cleaners, of which latter the centrifugal tar-extractor, illustrated in Fig. 174, page 451, has been found to be a simple and effective apparatus for cleaning any kind of bituminous gas and make it suitable for the gas-engine.

Anthracite Producers.—Minneapolis Suction Gas-Producer.—In Fig. 167 is shown a sectional view of a complete installation of an anthracite suction gas-producer of the type built by the

Minneapolis Steel and Machinery Co. The producer is fitted out with a pan-vaporizer, *W*, for generating the required quantity of steam. This vaporizer is built of steel as a ring-shaped pan, and it is placed on top of the furnace in such a manner as not to interfere with the poking of the fire through poke-openings, *P*,

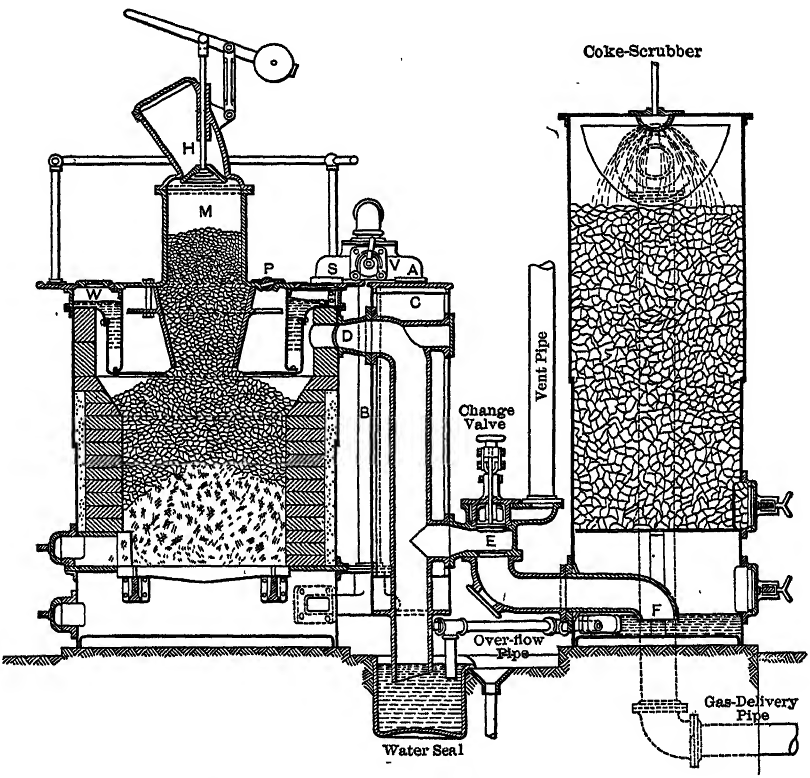


FIG. 167.—Suction Gas-Producer.

in line with the inside wall of the producer; it being important to be able effectively to remove, by poking, any cinders that tend to gather and build on to the walls. The gases escape around the outside of the waterleg of the pan to the gas-delivery pipe, *D*, which is carried down through an economizer, *C*, for pre-heating the air to be charged to the ash-pit. The sensible heat of the

escaping gases, is thus, well utilized, for generating vapor and for pre-heating the air.

A fuel-magazine, *M*, holding an ample supply of fuel, extends down through the centre of the water-pan, some distance, so as to bring the fuel-level to a proper height above the grate. The radiating heat from the top of the fuel-bed is taken up, partly by the fresh fuel-charge in the fuel-magazine, and partly by the inside surface of the water-pan. The heat-loss, due to radiation from the producer, becomes by this arrangement a minimum, and the producer top is kept cool.

A three-way valve, *V*, serves to regulate the proportion of air and steam furnished to the producer. The pre-heated air enters the valve at *A* and the steam at *S*, and by suitably adjusting the valve the proper proportion of air and steam will be delivered to the pipe, *B*, leading to the ash-pit. The cold air enters the economizer at *C*, and is carried down along one side of the gas discharge-pipe and up along its opposite side to the air admission, *A*, into the regulating valve.

E is a change-valve, which, when the fire is being blown up, is lowered so as to discharge the poor gases to the atmosphere, and when raised it discharges into the coke-scrubber, through the water-seal at *F*.

The coke-scrubber consists, as seen, simply of a steel-plate cylinder partially filled with coke, through which the gases pass slowly upward while water is delivered from a spray-nozzle above. Due to the large contact surface between the wet coke and the gases the latter will, in passing through the scrubber, become effectively cooled and cleaned from dust and tar.

From the coke-scrubber the gas is often carried directly to a small gas-tank, and from there to the engine. The object of the gas-tank is, partly, to separate and collect in it the water that may be carried along with the gas, but mainly to obviate in the producer, to some extent, the pulsations due to the periodical suction of the gas-engine piston.

Sometimes, there is placed in the gas-piping between the coke-scrubber and the engine a so-called dry scrubber, which apparatus is shown in the installation, Fig. 175. It consists of a steel-vessel

with a removable top, in which trays filled with shavings, or excelsior, are placed, through which the gas is filtered.

The trays of the purifier are, properly, made of wooden gratings, and the filtering material placed on them is often richly charged with iron turnings, which help to neutralize the acids formed by the oxidization of sulphurous fumes in the gases generated from fuels containing sulphur. The sulphur-compounds, H_2S and CS_2 in the gas are in themselves harmless to iron pipes and engine-parts, and act corroding first after oxidization to sulphuric acid. The conditions in the dry scrubber, the temperature, humidity, and presence of organic matter, are, however, very favorable for the oxidization of the sulphurous vapors, and it is in the apparatus itself the acids are formed for its own destruction. A heavy coating of iron oxide will help materially to preserve the metallic parts of the apparatus.

In order to effectively neutralize the acids, the trays in the dry scrubber are sometimes charged with a filtering material consisting of wood shavings, or excelsior, mixed heavily with iron oxide ore, so-called bog-iron ore. This material is generally used for the same purpose at gas-works and coke-oven plants.

Whenever a great percentage of sulphur is present in the gas any condensation of water-vapor in the engine should be carefully guarded against. The cooling, by jacket-water, of the exhaust valve-stem, piston or piston-rod and rod-packing should not be carried to such extent as to cause the deposit of vapor on these parts, and it has, in some cases, been found expedient not to use cold water for cooling the piston, piston-rod, and rod-packing.

The practice of injecting water into the exhaust pipe, in order to cool the exhaust gas, is very injurious to the piping, whenever traces of sulphur are present in the gas, and should, as far as producer-gas installations are concerned, never be indulged in.

"Olds" Suction Gas-Producer.—A suction gas-producer of a somewhat different type from the one just described, one with an outside vaporizer, is illustrated in Fig. 168, together with necessary cooling and cleaning apparatus. The vaporizer, *W*, is here built on the principle of a water-tube boiler, and it is attached to, and independent of, the producer proper. It consists of a set

of water-tubes, which connect a lower water-chamber with an upper steam-chamber, and a surrounding flue through which the hot producer gases pass; being, in passing, deflected to the right and to the left, laterally to the axes of the tubes, by two baffle plates. The producer-gases are collected at the centre of the fuel-level in the producer by the pipe, *P*, which conducts them in to the vaporizer, and from the latter they are delivered, through the change-valve, *C*, to the cooling scrubber.

The apparatus illustrated, which is a 300-horse-power producer of the Olds design, contains several new features of interest. The air for the combustion is taken from the outside in to a pre-heating duct, *D*, surrounding the upper part of the furnace-shell, is heated by being carried once around the shell, and is delivered to the ash-pit through the pipe *A*. With the air is, of course, charged also the required quantity of steam, which is furnished by the vaporizer, through the pipe *S*.

The producer is equipped with a water-sealed, revolvable, top, upon which a water-sealed charging-hopper is mounted, a proper distance off from the centre, so that by revolving the top the fuel can be charged all around the outside of the fuel-bed.

The grate is revolvable and presents a conical surface against the fuel-body, with a central, adequate opening at the bottom, through which ashes and cinders are freely discharged.

The change-valve, *C*, is particularly of interest, as it is one of a water-sealed type. The casing of this valve consists, as seen, of a cylindrical vessel holding a small amount of water, through which the vent-pipe, *V*, passes up, a short distance above the water-level. The valve proper is mainly an inverted cup, which is dropped over the orifice of the vent-pipe, whereby it effectively seals the gas-passage against the access of any air from without. When this cup-valve is raised out of the water-seal, by means of the rack and pinion shown in the illustration, it carries with it, in a surrounding ring-cup, a quantity of water, into which the orifice of the pipe leading to the scrubber dips, and becomes, thus, properly sealed against access for air. In its raised position the valve leaves the vent-orifice free for discharge from the producer to the atmosphere.

The cleaning apparatus consists, as seen, of a combination wet and dry scrubber. The gas is delivered to the wet scrubber through the pipe *B*, and passing up through the water-sprayed coke-body it becomes cooled and cleansed of dust, after which it

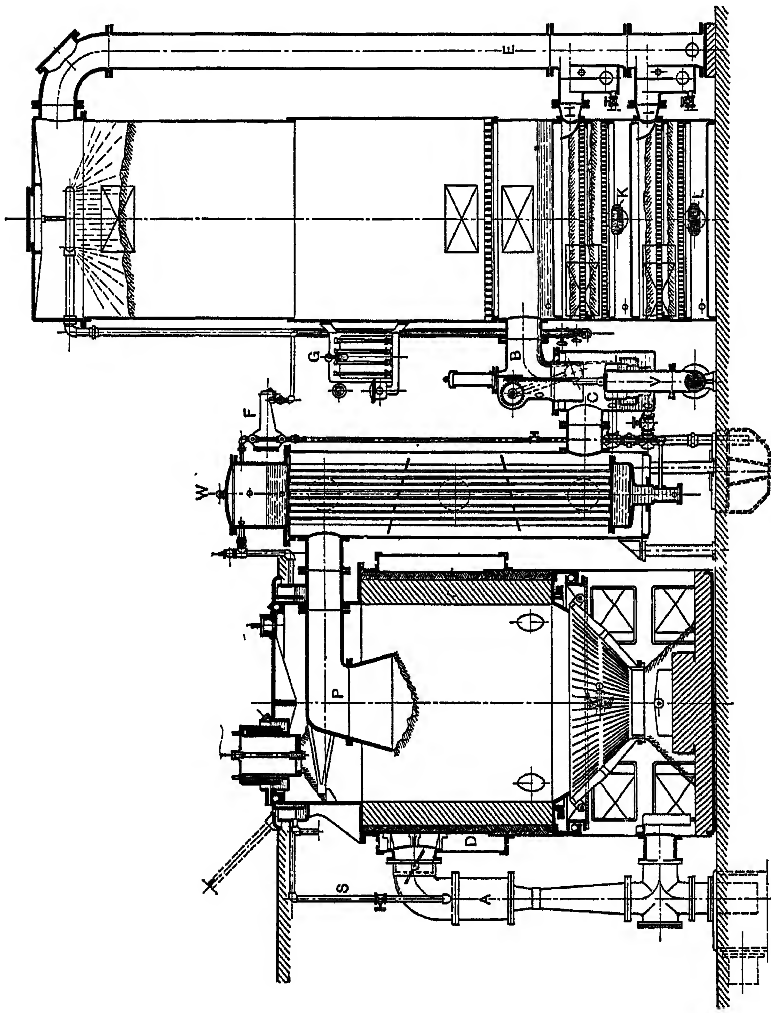


Fig. 168.—Suction Gas-Producer with Outside Vaporizer.

is passed, through the pipe *E*, in to a set of two dry scrubbers located in the base of the apparatus. The object of using two dry scrubbers is, simply, to double the filtering area through which

the total volume of gas must pass. The gas current divides itself, thus, between the upper and the lower filtering-chamber, passing in at *H* and *I*, and being delivered to the engine through the openings *K* and *L*.

G is a gauge-board with four water-gauges, each in order showing the gas-pressure, respectively, in the vaporizer, at the inlet side of the wet scrubber, at the outlet side of the wet scrubber, and at the outlet from the two dry scrubbers. By this means any obstruction in the gas-passages may readily be detected. A gauge-board of this kind, which makes it possible to ascertain quickly the pressure at different points of the gas-system, becomes, in connection with installations of some magnitude, a very necessary apparatus. A single water-gauge, piped up to the different points of the gas-system, with a small stop-cock in each branch may, however, answer the purpose.

F is a float-valve that controls, automatically, the water-level in the vaporizer.

Bituminous Producers.—The Water-Bottom Producer.—The water-bottom bituminous producer, Fig. 169, is a direct development from the original Siemens type. The grate for supporting the fuel is dispensed with, and the fuel-bed is made to rest on a column of ashes and refuse formed at the combustion; the whole fuel-column resting on the producer-bottom, which is made into a pan from which the refuse can very readily be removed, as required. The blast is carried to the centre of the column of ashes, and discharged by a distributing nozzle as evenly as possible over the full area of the bosh, some distance below the fuel-line. The producer is sealed by having the side-shell extended down into the ash-pan, which is luted with water.

This producer is used extensively for generating fuel-gas, and it may be said to be the standard type for fuel-gas. It is also, due to the fact that it involves a minimum amount of labor in its operation, used frequently for generating power-gas. It must then, however, be supplemented by effective cleaning apparatus for removing large quantities of tar.

More modern, bituminous producers are constructed and arranged with the idea of preventing the formation of any tar

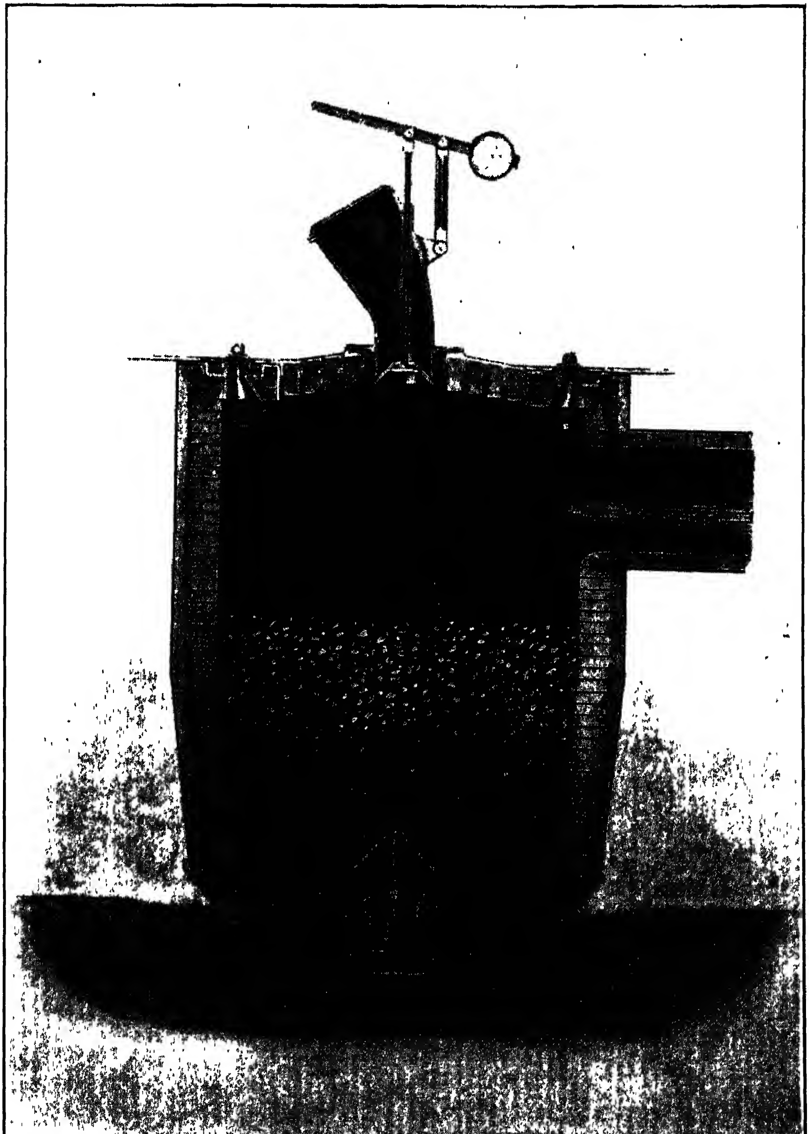


FIG. 169.—Water-Bottom Producer.

at the cooling of the gas. This is attempted by aiming to break up, in the furnace, into non-condensible gases, the hydrocarbons that form tar.

Tar is formed by the condensation of a variety of heavy hydrocarbons, many of which can readily be fixed into non-condensible gases by being heated to the required temperature. The new products which are formed at the decomposition of the hydrocarbons are hydrogen and carbon, or marsh-gas and carbon. Both these gases are stable and of high heating-value.

The Down-Draft Producer.—A producer built on this principle is the Loomis-Pettibone down-draft, double-furnace producer, a complete installation of which is shown in Fig. 170. In this producer the breaking up of the heavy hydrocarbons into incondensible gases is accomplished, in part at least, by carrying the gases which are distilled from the fresh fuel down through the incandescent coke in the lower part of the producer. The fuel will lose its volatile matter and become coked as it passes down into the furnace.

The producer proper consists, as will be seen, of two similar generators, 1 and 2, coupled to a common boiler, *F*, by means of the valves, *C* and *D*, so that they form one unit. A common exhauster, *E*, serves both generators for inducing the draft through their fuel-beds. Steam is introduced at the top of the generators and mingles with the air admitted through the top doors, *A* and *B*, which are open during the normal operation. The gases pass from the generators to the boiler for vaporizing the needed supply of steam, then, through the cooling and cleaning apparatus, to the gas-holder.

The fires may be regulated by passing, at regular intervals, a reversed current of steam through the fuel-beds for a fraction of a minute. This is effected by closing the openings, *A* and *B*, together with, for instance, the valve *C*, and admitting steam to the ash-pit of generator 1. The current, thus, passing up through the fuel-bed of this generator, then over to the top of generator 2 and down its fuel-bed, or *vice versa* if the valve *D* is closed and steam admitted to the ash-pit of generator 2.

The result of passing steam in this manner through the fuel-

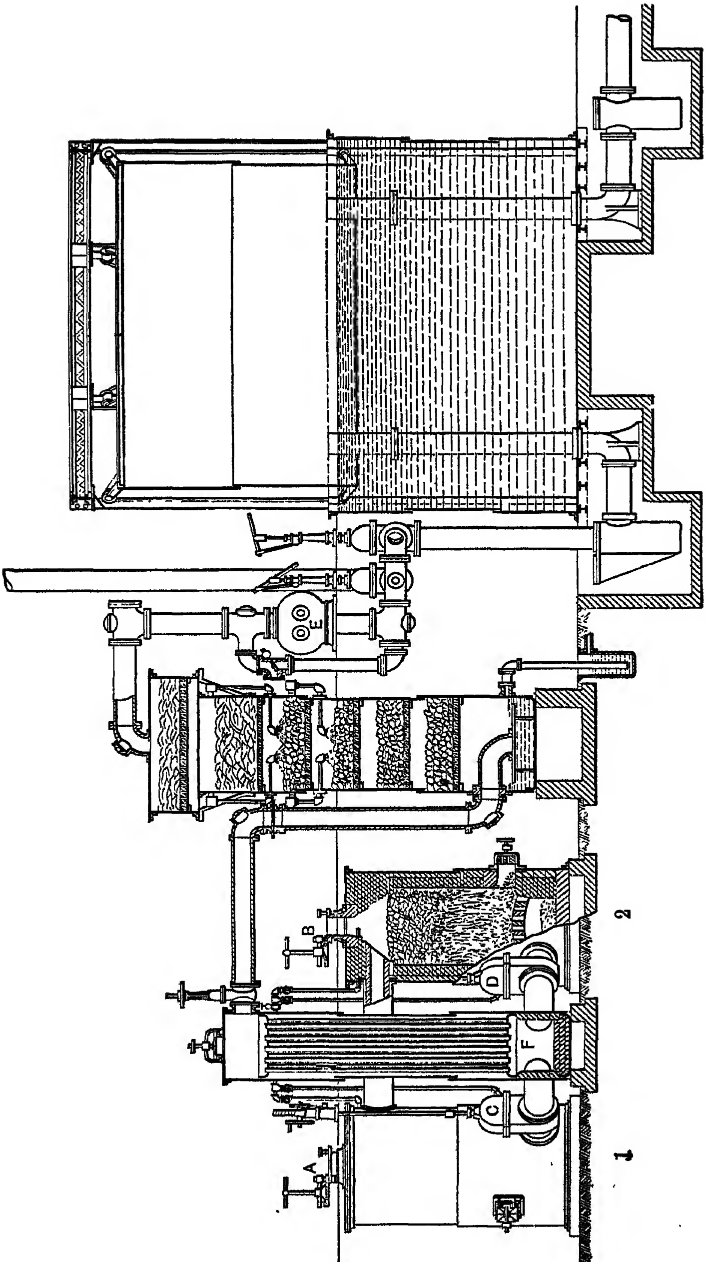


FIG. 170.—Loomis-Pettibone Gas-Generating Plant.

beds is the reduction of their temperatures, at the formation of water-gas. This gas being of a very high heating-value, it must be produced and mingled evenly with the gas normally generated, in order that any great fluctuation in the heating-value of the gas shall not be felt by the engine. To insure an even quality of gas, a gas-holder of some capacity is generally used in connection with this installation.

The gas-exhauster is connected to, and controlled by, the gas-holder, in such a manner that when the latter becomes filled with gas it shuts down the exhauster, thus stopping, for the time being, any production of gas.

The cleaning apparatus commonly used in connection with this type of producer, consists often only of a combination wet and dry scrubber, as shown in Fig. 170, supplemented by a sawdust purifier which is not shown.

The wet scrubber consists of a cylindrical steel-plate vessel provided with several trays heaped with coke over which water flows from a spray nozzle above. The gas is admitted at the bottom, and ascends slowly through the coke-bodies and is due to contact with a large surface of wet coke cooled and freed from tar and dust.

Double Zone Producers.—The Deutz double-zone producer, Fig. 171, is a combination of a down-draft and an up-draft producer. It contains two zones of incandescent fuel. Air and steam are admitted into the upper part of the producer and the draft is downward. The hydrocarbons that are distilled off from the fresh fuel are thus drawn down through a bed of incandescent coke below, and fixed into a non-condensible gas, which is taken out through a delivery-pipe about midway of the height of the producer. At the same time there is another current from the ash-pit up through the lower zone of the producer. The main object with this latter current is to consume that part of the fuel that otherwise would pass out of the upper zone unconsumed. It helps also to maintain the upper zone in a high state of incandescence, thereby contributing to the thorough decomposition of most of the heavy hydrocarbons and to the reduction of the carbon dioxide gases formed in the upper part

of the producer. This producer appears to have many good points, as far as the production of a stable gas is concerned, but, in common with many of the bituminous producers, it requires excessive labor to keep the fires in proper condition, particularly

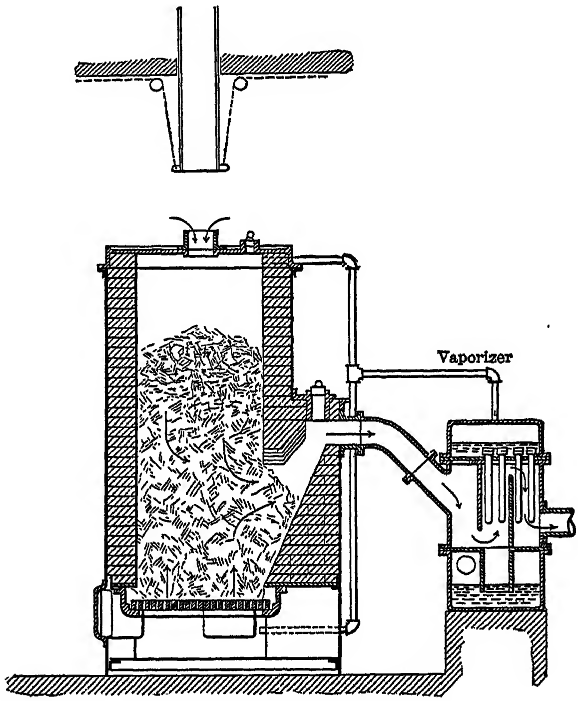


FIG. 171.—Deutz Double-Zone Producer.

if the fuel is not absolutely non-coking, and it is not very readily cleaned. In places in Europe it has, however, been very successfully used on lignites and briquetted peat.

Lignite and Peat.—The subject of lignite fuels for power purposes is at present of the very greatest interest, there being in the northwestern States, Dakota, Wyoming, and Montana, immense fields of this fuel available for use in the power gas-producer. The fuel is, however, of such a low grade that it cannot, under present conditions, be transported any distance with economy, and it becomes therefore mainly suitable only for home

consumption. It will, on this account, be necessary to locate the gas-producer plant conveniently to the fuel-supply.

In the following table are given the composition and heating-value of samples of lignite from different localities, and, for the purpose of comparison, the analyses of two other fuels, wood and peat, that are occasionally used in the producer.

The heating-values of wood and peat are those of the perfectly dried fuels, the moisture in the air-dried samples having been allowed for.

The heating-values of the American lignites are those given by the analyses of air-dried samples, and, the second figure, those corresponding to perfectly dried samples.

When drying in the air, the lignites lose from 10 to 30 per cent of their original water contents.

	Wood. Average.	Peat. Average.	Lignite.	
			Alameda Co., California.	Stark Co., N. Dakota.
Moisture in air-dried fuel	20	22	18.5	32.6
Volatile matter	....	....	35.3	29.2
Fixed carbon	....	....	30.7	26.8
Ash	....	....	15 5	11.4
Ash	1.5	6 0	15.5	11.4
Sulphur	....	....	3.0	3.5
Hydrogen	6.2	6.0	5.9	6.2
Carbon	50.0	56.0	47.4	39.5
Nitrogen	1.0	1.0	.7	.5
Oxygen	41.3	31.0	27.5	38.9
Calorific value as per analyses	....	....	8105	6970
Calorific value, moisture allowed for	7000	10200	10020	10800

An endurance test of a Deutz two-zone, down-and-up draft producer on lignite fuel, lasting 320 hours, showed a fuel consumption per brake horse-power of 1.46 pounds.

An analysis of the fuel gave:

A calorific value of 8,500 B.T.U.

Fixed carbon	35.0	per cent.
Volatile matter	45.0	" "
Ash	4.8	" "
Water	15.2	" "

The resulting gas showed no appreciable amount of tar.

Another test of this producer showed at heavy load (100 B.H.P.), a fuel consumption of 1.1 pound.

The analysis of the fuel gave:

Fixed carbon	42.0	per cent.
Volatile matter	31.2	" "
Ash	4.4	" "
Water	22.45	" "

The Jahn Producer.—The Jahn producer, Figs. 172 and 173, which in Germany is working very successfully on any grade of cheap fuel, consists of four up-draft furnaces or retorts connected by means of valves, in such a manner that the current of gas can be shifted from one furnace of the set in to any other one of the same set. The furnaces are charged at regular intervals, starting with number 1, next number 2, and so on, in succession. The gases from the latest charged furnace, number 4, which to a great extent consist of hydrocarbons and moisture distilled off from the fresh fuel, are drawn through the fuel-beds of the second and third units. By this means, it is quite apparent, the gases, when they leave the system, must be of a highly stable quality, they having been passed through the incandescent coke-beds of the second and third units. Furnace number 1 of the set is, in the mean time, switched out of the system for being cleaned, re-charged and put in readiness to take the place of unit number 4, in due time.

The general construction of the producer may be studied by reference to Figs. 172 and 173. The upper half of Fig. 172 represents a plan view looking from the top of two furnaces, and the lower half is sections taken through the furnaces, at *C C* and *D D*. The left-hand half of Fig. 173 represents a vertical section

taken on the line $A\ A$, and the right-hand half is a section taken, on the bias, through the line $B\ B$.

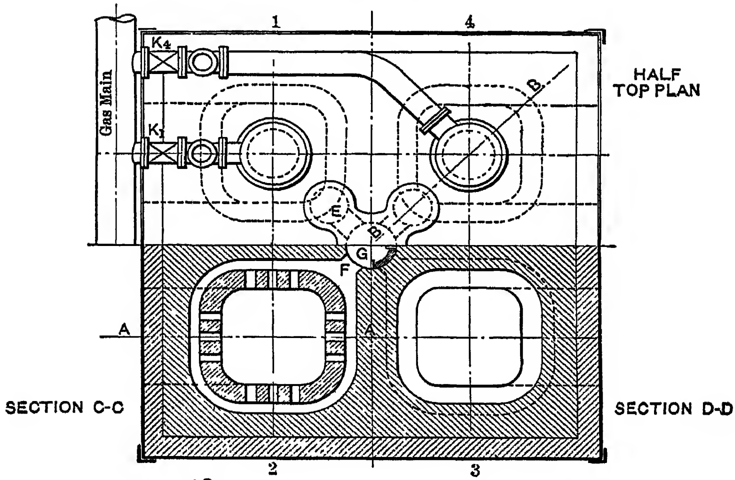
It will be seen that each individual furnace communicates, through ports E and F, with a central gas-flue, G, respectively, above the main part of the furnace and at the bottom of the furnace. These passages can be opened and closed according to requirements by valves such as I and J, which are all operated from the charging floor. The main shafts of the furnaces terminate, above the charging floor, in sealed hoppers, H, through which the fuel is charged, and through which also the fixed producer-gases are conducted to the gas-main.

As each one of the hoppers, H, will, in turn, be opened at times when its furnace is being cleaned and re-charged, there must, of course, be provided a gas-valve in the pipes communicating between the hoppers and the gas-main. These valves, for furnaces 1 and 4, are shown at K_1 and K_4. It will be necessary, at times when the furnaces are first started up, to get rid of some gas too poor for being used, and for that purpose there is also provided for each furnace a vent-valve, which is shown at M.

The gas which is fixed and ready for the engines is, by means of an exhauster, drawn from the furnaces through suitable cooling and cleaning apparatus, and delivered to a gas-holder.

Assuming that the furnaces have been put in normal operation, and that furnace number 4 has been switched out of the system for cleaning and re-charging. The valves I and J are then, as shown in the drawing, closed, as is also the valve K_4 in the gas-pipe leading from hopper of furnace 4. This hopper, as well as the fire- and ash-doors of furnace 4, can then be opened to facilitate the cleaning of the furnace, because the furnace is isolated from the rest of the system. When ready to be switched in to the system, the hopper and fire-door of furnace 4 are closed and the valve I opened; the valve J and the gas-valve, K_4, remaining closed. The draft current will, thus, be: From the ash-pit through the fuel-bed of furnace 4, then through the port E in to the central gas-flue, G. From there it will pass through the passage F in to the incandescent coke-beds of furnaces 2 and 3 to be drawn from their hoppers to the general gas-main;

FIG. 172.



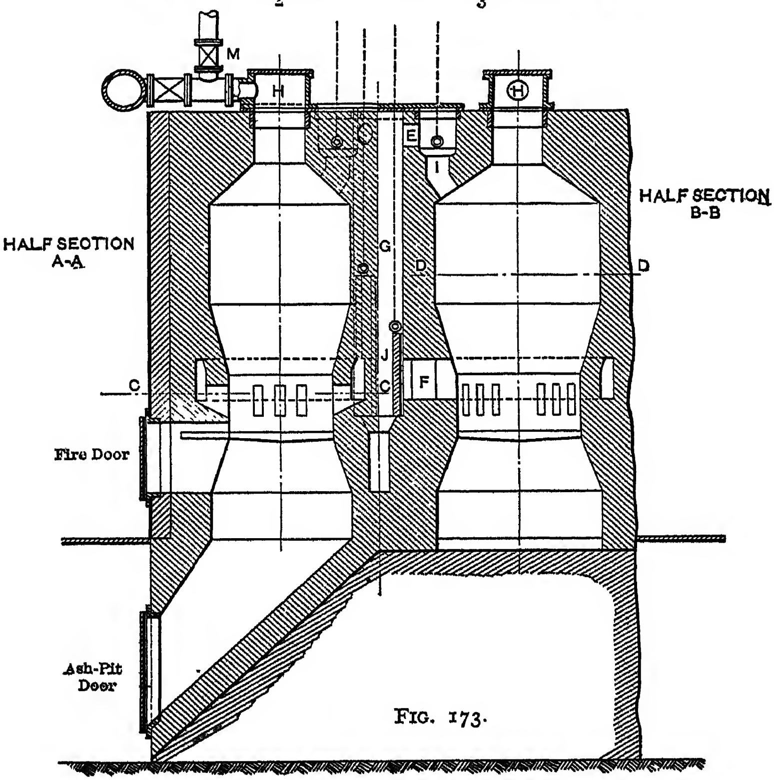


FIG. 173.

Jahn Producer.

furnace 1 being supposed to have been in the mean time, switched out of the system to be cleaned.

The furnaces are built of capacities to hold a charge of three to four tons, and may be arranged as many as required in one group; each four furnaces being one set. The brickwork must be laid in a substantial manner, and the whole group of furnaces is encased in a steel casing.

A successful power-gas plant, containing several of the Jahn producers, has been in use, since 1902, at the Van der Heydt coal mines in Germany, the fuel being refuse containing only 20 per cent coal. The producer seems to possess, in a promising manner, all the requirements for a continuous production of a fixed gas from low-grade fuels, excepting one—convenience for poking the fires to keep the fuel-beds open, and it has been learned that coal that has the tendency to coke is unfit for use in this producer.

Gas-Washers.—A centrifugal gas-washer of efficient type is shown in Fig. 174. It consists of a horizontal drum fitted with radial vanes, which is journaled in a gas-tight steel casing also carrying vanes that clear those on the drum by a small margin; one-half to three-quarters of an inch. The drum is revolved at a suitable speed; and the gas, after having been cooled in some kind of cooling apparatus, is admitted to the washer through the pipe, *P*, and in passing between the stationary and revolving vanes of the apparatus is thoroughly mixed with a spray of water admitted through the pipe *W*. The tar-products and water are beaten up to an emulsion which passes off along the shell of the casing to the bottom of the washer; collecting in the lower settling-chamber, and passing out at *D* to the drain-tank, *T*.

The current through the apparatus is impelled by the vanes *I* at the outlet end of the revolving drum, and the gas leaves the discharge-chamber, *E*, at a slight pressure. If the pressure in the discharge-chamber should, due to a decreased gas consumption, increase a small amount above that contemplated, then the excess of gas will become by-passed through the water-seal at *S* and be returned through the ports *R* to the inlet side of the washer. There is provided a water-seal also at the bottom of the

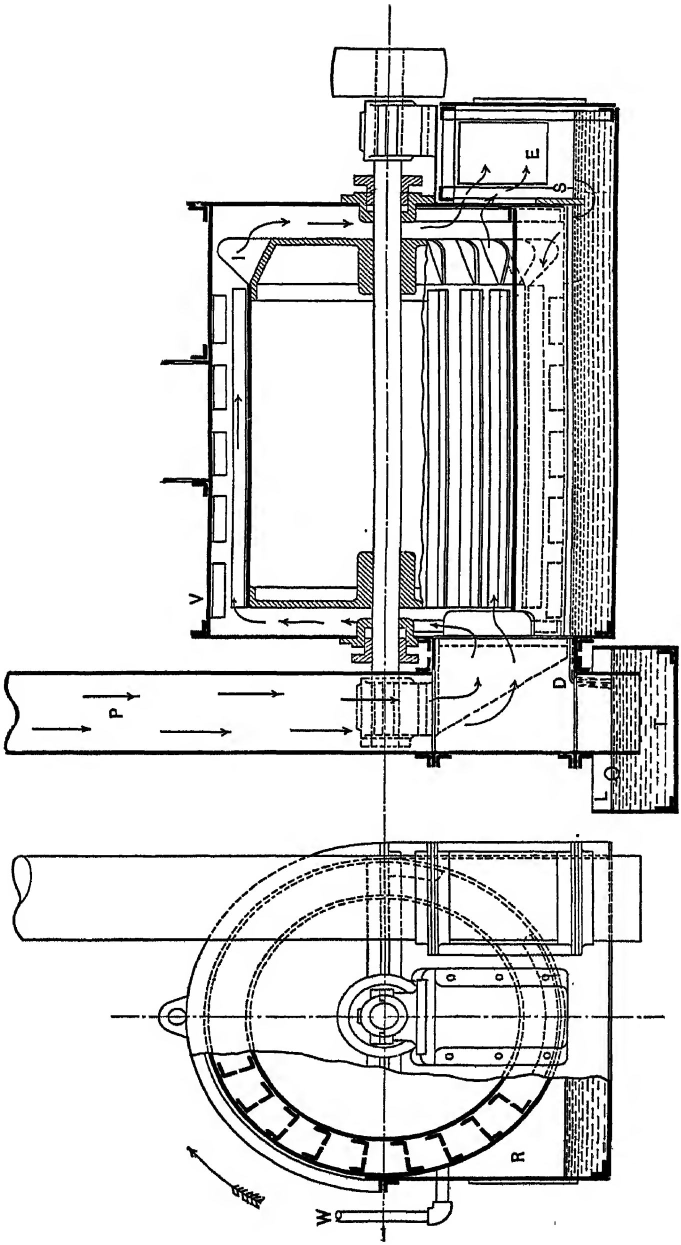


FIG. 174.—Centrifugal Gas-Washer.

inlet gas-pipe to afford a relief for the pressure in case an explosion should occur in the apparatus.

The light tar-products will be drained off from the drain-tank at *L*, whereas heavy products will be collected and drained from the bottom of the tank, or from the bottom of the settling-chamber of the washer. An effective apparatus of this type will readily remove the tar in the gas to such an extent that the impurities remaining shall not average more than 0.015 to 0.02 grains per cubic foot; the gas thus being amply clean for any purpose for which it may be used.

A washer handling the gas from a 600 horse-power producer would normally require 15 to 20 horse-power, but, as some overload capacity will be required in starting, a 25 horse-power engine or motor would suitably be provided. The water-consumption will not exceed two gallons per horse-power per hour.

Capacity of Producers.—The rate at which coals may properly be gasified in the gas-producer is at present a more or less unsettled question, due to the variety in the character of the coals generally used. It has frequently been held by German authorities, that in an anthracite producer seven to nine square inches of sectional area of the fuel-column would be a proper allowance per horse-power, but the practice with American anthracite coal has shown that ten to sixteen square inches give far better results on a somewhat steady load. When the load-conditions are such that the engine may run very light for a considerable length of time (say, less than one-quarter of full load for about twenty minutes), and then the full load be suddenly thrown on, and if such changes are liable to occur repeatedly, then it is advisable to install a relatively small producer. Since the large producer is unnecessarily large at light loads, it will, if the vapor-supply is not carefully controlled, become cooled down during the light load to such an extent that when the heavy load comes on it may not be in a condition to respond with the required supply of gas. The smaller producer, on the other hand, will not, under reasonable short intervals of heavy load, have time to become heated enough to give trouble on account of a clinkering fuel, and will maintain better a normal condition during light-load

periods. When the load is steady a large producer is, however, to be preferred.

When a producer is used for generating gas for power purposes its capacity is generally expressed in horse-power, although it would be more correct to express it by the number of pounds of coal it can gasify per hour. If the fuel-consumption be counted $1\frac{1}{4}$ pound of coal per horse-power, then 10 to 16 square inches sectional area per horse-power would correspond to a gasification of 18 and 11 pounds of fuel per square foot fuel-area per hour.

In a bituminous producer good gas can be generated from suitable coal at the rate of 20 pounds of fuel, or more, per square foot, but with less suitable fuels the gasification should not be driven at a higher rate than 10 pounds per square foot; the best rate for gasification being very much dependent on the quality of the fuel.

The reaction in the producer whereby good gas is produced depends on the amount of hot fuel-surface exposed to the action of the gases passing through the fuel-bed, its temperature, and the speed with which the gases pass. If the current is forced through at a high rate of speed the temperature of the fuel-bed must be high in order to reduce the $C\,O_2$-gas properly, and the hot zone becomes high, wherefore the gases pass off hot. If, in such a case, the fuel is of a good quality, containing a low percentage of ash and refuse of a high fusing-point, then the production of good gas will proceed with no other inconvenience than that there will be incurred some loss due to the excessive amount of sensible heat carried off by the gases. On the other hand, should the fuel be of a low grade, containing a great amount of highly fusible refuse, then great inconvenience will be experienced on account of the formation of clinkers that impede the even flow of the gases over the whole area of the fuel-bed. The current will seek channels forming through the fuel-bed and gas of a poor quality will escape. A continuous poking of the fire to remove clinkers must in this case be resorted to, in order to keep up the gas-production.

With low-grade fuels, the operation of the producer, at the rate of 10 pounds of fuel per square foot fuel-area and at a tem-

perature of the fuel-bed of 1,800 to 2,000° F., may be continued without any great inconvenience from the fusing of the ash, whereas at an increase of 25 per cent of the output the temperature of the furnace may rise to 2,200 degrees and cause a great expenditure of labor to keep the operation going.

With regard to low-grade fuels containing a great amount of highly fusible ash, the nearest to a correct statement as to the rate at which it should be gasified may, perhaps, be to say, that the proper rate is only that which involves the least amount of labor for producing a gas of good quality.

Size of the Gas-Outlet Pipe.—The following may serve as a guide for determining the size of the gas-outlet pipe of a power-gas producer. There is generated, in the anthracite suction producer, about 5½ pounds of gas per pound of coal, or, in volume, on an average, 80 cubic feet at 62° F. Assuming the gas to leave the producer at 900 degrees, its volume at that temperature will be 208 cubic feet per pound of coal, or, per pound of coal gasified per hour, there will be discharged 0.06 cubic feet per second.

Limiting the mean speed of the gases through the discharge-opening to 30 feet per second, the required area of the opening becomes 0.28 square inches per pound of coal gasified per hour. There is generally figured, as a safe allowance, 1¼ pound of anthracite fuel per brake horse-power per hour, and, hence, the required discharge opening per horse-power would be 0.35 square inches.

For the type of anthracite producer shown in Figs. 167 and 168 this area of discharge pipe is ample.

The gas from a bituminous producer is often of a higher temperature than 900 degrees, it being less thoroughly cooled by the fresh fuel-charge and vaporizer, but, on the other hand, the volume of the gas generated per pound of fuel is somewhat less than that of anthracite gas. An area of the discharge opening of 0.28 square inch per pound of fuel gasified per hour, or 0.35 square inch per horse-power, will therefore be ample even for the bituminous power gas-producer. When cooled, the volume of the gas is less than one-half the volume at 900 degrees, wherefore

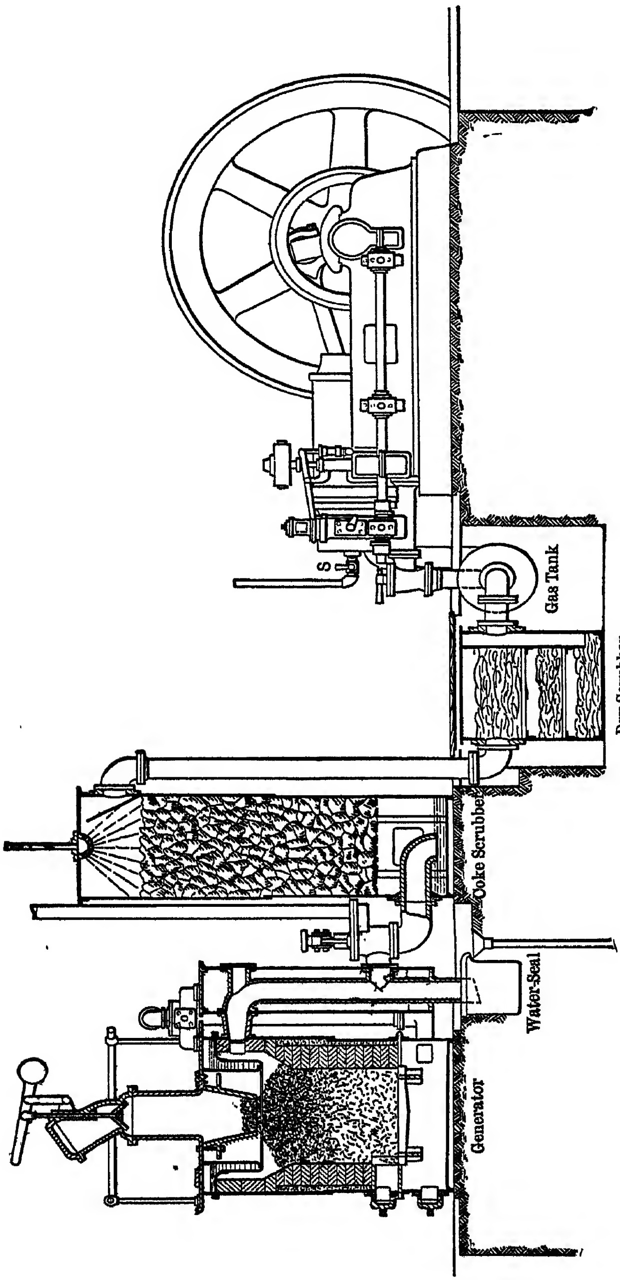


FIG. 175.—Suction-Gas Power Installation.

the pipe carrying the cool gas can, consistently, be smaller than the hot gas-pipe.

For a fuel-gas producer the discharge pipe would be larger than the proportions given, which apply for power-gas producers. Figuring the maximum capacity of a fuel-gas producer 18 pounds of fuel per square foot area, and the discharge-opening 0.5 square inch per pound of fuel gasified, the discharge-pipe would be nine square inches in area for every square foot area of the producer, or at a ratio of 1 to 16. The diameter of the discharge-pipe, therefore, one-quarter of the diameter of the producer.

Producer and Gas-Engine Installation.—Fig. 175 illustrates a complete installation of a suction gas-producer and engine. In the way of cleaning apparatus the installation contains the usual wet scrubber (coke-scrubber), and a dry scrubber which, in order to economize on space, is placed below the floor-line. The engine is started by means of compressed air which is delivered from an overhead air-tank to the starting-valve, *S*. An air-pressure of 120 to 140 pounds is commonly used. The size of the air-compressor, usually driven by a small gasoline- or gas-engine, and the size of the air-tank, should be suited to each other so that the time for storing up the pressure does not become excessive.

For all single engines of Table XXV a 3 x 4 single-acting, two-cylinder compressor running 200 to 250 turns will be adequate, and the same compressor will do for twin or tandem engines up to a size of 17 x 28. For any twin or tandem engine, referred to in the table, above that size a 3¾ x 5 compressor will be better.

The complete installation must also include a pump for circulating the cooling-water through the jacket of the engine. The required size of such a pump may be determined according to the method given at page 297.

APPENDIX

Gas-Engine Tests.—In the following are recorded, in detail, a few gas-engine tests of acknowledged high efficiencies, comprising various kinds of fuels. The main data relating to these tests have been included in Table XXXI, and below will be found, in carefully tabulated form, the detail data of interest as given by the experimenter, but transcribed to the English system of weights and measures.

TEST OF A 500-HORSE-POWER BORSIG-OECHELHAEUSER ENGINE on coke-oven gas, at Silesia, Germany. Test by Prof. E. Meyer of Charlottenburg, Oct. 26, 1903.

The main dimensions of the engine, which is of the two-cycle type shown in Figs. 134*a* and 134*b*, are:

WORKING CYLINDER.		
Diameter	26.6	inches
Length of stroke	37.5	inches
BLOWING TUB.		
Diameter	65.0	inches
Length of stroke	37.5	inches
Diameter of piston-rod	6.0	inches
DOUBLE-ACTING AIR PUMP.		
Diameter of piston	45.0	inches
Length of stroke	19.6	inches
Diameter of piston-rod, front	3.54	inches
Diameter of piston-rod, rear	2.75	inches
SINGLE-ACTING GAS PUMP.		
Diameter of piston	23.2	inches
Length of stroke	19.6	inches

The composition of the gas used is shown by the following table:

Analyses of Coke-Oven Gas.

TEST NUMBER	I.	II.	III.
Time	10 a.m.	4 p.m.	10 a.m.
PER CENT HYDROGEN, H	42.00	48.08	43.80
Carbon monoxide, CO	11.84	10.60	10.20
Heavy Hydrocarbons	2.63	1.80	2.10
Methane, CH_4	19.73	18.43	20.30
Nitrogen, N	18.69	15.89	17.90
Carbon dioxide, CO_2	4.91	4.90	5.30
Oxygen, O	0.20	0.30	0.40

TABLE XXXII.

Data from Test of October 26, 1903, of a 500 H. P. Borsig-Oechelhaeuser Engine, Working on Coke-Oven Gas.

		VIII.	IX.	X.	VI.	VII.
Number of test		20	15	40	15	20
Duration of test—minutes		103.0	107.0	106.1	108.2	107.4
Speed of engine—revolutions per minute		75.1	73.8	69.3	62.3	62.0
Working cylinder	M.E.P.—lbs per sq. in.					
	Total indicated work = I.H.P.t	810	827	769	705	697
Blowing cylinder	Total indicated work, or brake horse-power of the engine = B.H.P.	616.2	627	575	488	474
Air pump	M.E.P., in front of piston—lbs.	5.09	5.38	5.12	5.56	6.09
	M.E.P., back of piston—lbs.	3.36	3.58	3.41	3.73	3.94
	Indicated H.P. Consumed = Wa	68.3	75.2	71.1	79.0	84.5
Gas pump	M.E.P.—lbs.	3.53	3.50	3.58	3.73	3.84
	Indicated H.P. consumed = Wg	7.7	7.8	7.9	8.5	8.7
Total indicated H.P. consumed by charging pumps = Wa + Wg		76.0	83.0	79.1	87.5	93.2
Net indicated work of engine = I.H.P.n = I.H.P.t − Wa − Wg		734	744	690	617	603
$\frac{\text{Total pump work}}{\text{Net ind. work}} \times 100 = \frac{W_a + W_g}{\text{I.H.P.}_n} \times 100$—%		10.3	11.1	11.4	14.2	15.5
Total efficiency between working cylinder and blower = $\frac{\text{B H.P.}}{\text{I.H.P.}_t} \times 100$—%		76.1	75.8	74.8	69.2	68.0
Mechanical efficiency of engine = $\frac{\text{B.H.P.}}{\text{I.H.P.}_n} \times 100$—%		83.9	84.2	83.3	79.1	78.6
Work consumed in friction in the engine = I.H.P.n − B.H P.		117.8	117.	115.	129.	129.
Gas of 32°F. and 760 m/m	consumed per hour—*c. f.*	13505	13951	13198	12100	11800
	lower heating-value B.T.U. per *c. f.*	400.1	394.5	383.3	394.5	397.9
Heat consumption	Total per hour—B.T.U.	5,505,000	5,504,000	5,060,000	4,774,000	4,694,000
	per total I.H.P.—B.T.U.	6679	6639	6599	6760	6720
	per net I.H.P.—B.T.U.	7364	7404	7324	7726	7766
	per brake H.P.—B.T.U.	8772	8772	8772	9778	9899

This engine test is notable, not only on account of the high efficiency it records, but also on account of the unusually low average heating-value per cubic foot of actual charge.

Figuring the capacity per unit volume of the cylinder at par with that of a non-scavenging engine (the suction displacement-volume thus equal to the volume of the actual charge) we find:
The suction displacement-volume per revolution to be 24.1 cubic feet;
The suction displacement-volume, per minute, of test X, 2557 cubic feet;
The total indicated power of test X is 769 I.H.P.;
Hence, the suction displacement per ind. horse-power is D = 3.32 cubic feet.
The thermal efficiency of test X is $Efy = 0.385$ with respect to the indicated output.

If the above quantities are inserted in formula 52 we obtain:

$$\frac{H}{\frac{V_a}{V_o}\left(xa + 1\right)} = \frac{42.42}{Efy\ D} = \frac{42.42}{0.385 \times 3.22} = 33.2.$$

Thus, referring to test X, the heating-value per cubic foot of actual charge is 33.2 B.T.U. only; and with respect to the other tests of the series it will be found also to approximate this figure.

The assumptions made in Chapter VI, with respect to the normal charge were the following:
The excess air is 15 per cent of that theoretically required, the pressure of the charge at completed suction-stroke is 13.2 pounds absolute, and its temperature is 120° F. Under these conditions the normal charge would be of a heating-value of 61 B.T.U. per cubic foot. Hence, the preceding test would indicate that for obtaining the best efficiency the required excess air-charge should be much greater than the assumed 15 per cent allowed in the normal charge. It is true that many efficiency tests on illuminating- or coke-oven gas have given excellent results with an excess dilution of 30 to 50 per cent above that theoretically required, but, then again, many tests show poor results when a too highly diluted charge has been used, and it is exceptional that a very

high efficiency is obtained with a charge diluted, as in the above test, to approximately twice the volume of that which in the previous has been designated the normal charge.

As an explanation of the high efficiency obtained in the above case in spite of the highly diluted charge used, the circumstance that the hot neutrals are not fully expelled from the cylinder would probably be quoted; and that they are separated from the fresh charge, which therefore, actually, contains essentially less air than appears from the above computation. In this case the neutrals remaining in the cylinder serve in the main only to reduce the effective cylinder-volume.

For the sake of comparison with the above test, it will be of interest to make some deductions from a test of another two-cycle engine of high economy.

The test of a Koerting engine at Hanover, recorded in Table XXXI, gave on producer-gas an efficiency $Efy = 0.34$. The cylinder dimensions are 21.6×37.7 inches, and the speed of the engine was 101 revolutions per minute.

Assuming the piston-rod to be of a diameter of 5 inches, we obtain:

The suction displacement per stroke 7.55 cubic feet, and the suction displacement per minute 1525 cubic feet.

The work generated was 481 I.H.P., and, hence,

the suction displacement per indicated horse-power per minute 3.17 cubic feet.

The heating-value of the actual charge becomes, therefore, per cubic feet:

$$\frac{H}{\frac{V_a}{V_o}\left(x\,a + 1\right)} = \frac{42.42}{0.34 \times 3.17} = 40 \text{ B.T.U.}$$

The low heating-value of the gas used being 129 B.T.U. per cubic foot, the heating-value per cubic foot of normal charge would probably be in the neighborhood of 45 B.T.U.

In this case we have, therefore, a two-cycle engine which gives a high efficiency, using a charge much less diluted than in the case of the previous test.

In the case of the test of the Premier engine at Winnington, also recorded in Table XXXI, a thermal efficiency referred to the indicated output of 0.337 was obtained. This engine, of four-cycle scavenging type, has a suction displacement of 10.61 cubic feet per revolution, which is equivalent to a suction displacement, at the test, of 2.78 cubic feet per indicated horse-power per minute.

The heating-value per cubic foot of charge was, accordingly, at the test 46 B.T.U.; whereas the heating-value of the so-called normal charge (the gas being of 144 B.T.U.) would probably approximate 48 B.T.U.

Again, the test of the double-acting Nuernberg blast-furnace gas-engine at Rombach, recorded in Table XXXI, gave a thermal efficiency $Efy = 0.339$, and the suction displacement per indicated horse-power per minute was $D = 3.1$ cubic feet. The heating-value per cubic foot of charge, accordingly, 40 B.T.U., whereas the heating-value of the so-called normal charge (the gas being of 88 B.T.U.) would probably be somewhat less than 40 B.T.U.

From the above deductions it will appear, thus, that it is not generally necessary for obtaining a good economy that the charge should contain a very great excess of air, though it is possible that an apparently highly diluted charge may give a very ample efficiency, if the neutrals in the cylinder can be kept separated from the actual charge.

Test of a 300 Horse-power High-speed Diesel Engine at the Augsburg Works of the Diesel Co. The test made by Chr. Eberly, of Munich, and recorded at page 180 of the Zeitschrift des Vereins Deutscher Ingenieure, February 1, 1908.

The fuel used was Galician crude oil of the following composition:

Carbon	86.41 per cent
Hydrogen	12.66 per cent
Sulphur	0.85 per cent
Oxygen and nitrogen	0.08 per cent
	100.00

Lower heating value, 18,130 B.T.U. per pound.

TABLE

Data from Test of a 300-

Number of test	1	2
Duration of test—hours	1.16	1.18
Speed of engine—revolutions per minute	256.8	306.6
Indicated power of working cylinder less work consumed by air-pump—I.H.P.	239	292
Brake horse-power of engine—B.H.P.	197	233
Mech. efficiency = $\frac{\text{Brake H.P.}}{\text{Ind. H.P. of work. cyl.} - \text{Air-pump work.}}$	82.5	80.0
Fuel consumption per hour: per ind. H.P. neglecting air-pump work—lbs	0.322	0.316
Fuel consumption per hour: per brake H.P.—lbs	0.420	0.426
Fuel consumption per hour: per brake H.P. referred to fuel of 17600 B.T.U. per pound, lower value—lbs	0.431	0.437
Cooling water used per brake H. P. per hour—lbs	69.5	58.5
Temperature of cooling water: at inlet—°F	55	55
Temperature of cooling water: at outlet—°F	93	100
Temperature of exhaust gases—°F	624	696
Analyses of exhaust gases: contents CO_2-gas—%	6.8	8.3
Analyses of exhaust gases: contents oxygen—%	8.4	......
Heating-value of the fuel B.T.U. per lbs.	18126	......
Heating-value consumed: per ind. H.P.—B.T.U.	5842	5738
Heating-value consumed: per brake H.P.—B.T.U.	7617	7718

Heat Distribution per One Pound of the Fuel.	B.T.U.	%	B.T.U.	%
Converted into indicated work	7893	43.5	8037	44.3
Converted into brake horse-power	6057	33.4	5967	32.9
Friction, and work consumed by the air-, oil-, and water pumps	1836	10.0	2070	11.4
Dissipated into the cooling water	6210	34.3	6066	33.5
Carried off by the exhaust gases	4374	24.1	4140	22.9
Discrepancy	-351	-1.9	-117	-0.7
Total = lower heating value of the fuel	18126	....	18126	....

XXXIII.

Horse-Power Diesel Engine.

3	5	7	8	9	10	11	12
0.83	0.85	0.53	0.42	0.49	0.46	0.53	0.60
402.4	497.4	301.1	247.4	400.1	488.1	400.5	508.1
386	402	181	146	221	246	393	....
294	335	126	106	156	165	322	390
76.2	83.3	69.6	72.6	70.6	67.2	81.8	....
0.306	0.337	0.284	0.292	0.302	0.334	0.328	....
0.435	0.454	0.453	0.448	0.483	0.560	0.439	0.460
0.447	0.466	0.464	0.460	0.496	0.574	0.451	0.483
58.3	57.4	....	....	....	....	....	....
55	55	....	....	....	....	....	....
99	104	....	....	....	....	....	....
734	826	432	401	496	617	762	....
8.4	9.4	4.5	4.4	4.6	4.8	9.6	....
8.0	7.0	13.8	14.0	13.2	12.6	8.4	....
....	....	....	....	....	....	....	....
5553	6121	5142	5292	5477	6064	5963	....
7903	8225	8205	8124	8772	10152	7963	8531

B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%
8298	45.8	7524	41.5	8962	49.4	8712	48.0	8415	46.4	7605	41.9	7731	42.6	....	....
5841	32.2	5607	30.9	5625	31.0	5679	31.3	5265	29.0	4536	25.0	5787	31.9	5409	29.8
2457	13.6	1917	10.6	3337	18.4	3033	16.7	3150	17.4	3069	16.9	1944	10.7	....	....
5760	31.8	6138	33.8	....	....	....	....	....	....	....	....	....	....	....	....
4338	23.9	4482	24.8	....	....	....	....	....	....	....	....	....	....	....	....
−270	−1.5	−18	−0.1	....	....	....	....	....	....	....	....	....	....	....	....
18126	....	18126	....	....	....	....	....	....	....	....	....	....	....	....	....

Table XXXIII contains the results of the principal tests of the complete series, and in Fig. 177 are represented the fuel-consumption curves for four different speeds of the engine.

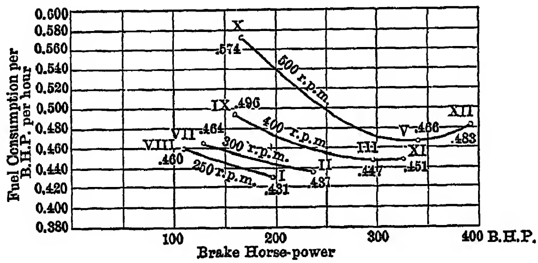


FIG. 177.—Fuel-Consumption Curves of Diesel 500 H.P. High-Speed Oil-Engine.

DUTY TEST OF A WESTINGHOUSE HORIZONTAL 500 HORSE-POWER DOUBLE-ACTING TANDEM ENGINE at the Plant of the Norton Co., Worcester, Mass.

Test made June 24 to 26, 1907, by G. J. Alden and J. R. Bibbins.

Cylinders: 23½ inches diameter,
33 inches stroke.

Bituminous gas-producer of the Loomis-Pettibone down-draft type.

A series of fuel-consumption tests on varying loads were made by observing the drop of the large gas-holder during the time each test was in progress. These tests are recorded in Table XXXIV, and graphically in Fig. 178.

There was also carried out a 51-hour test under average factory conditions, the general results of which are recorded in Table XXXV.

The mechanical efficiency of the engine was determined by comparing the results of 72 sets of indicator cards with the electrical output at the time the cards were taken; the generator efficiency having been carefully determined. The average mechanical efficiency was 83.5 per cent. Clearfield bituminous coal was used, the analysis of which averaged:

Volatile matter	19.87
Fixed carbon	73.71
Moisture	0.87
Ash	5.54
Sulphur	0.83
Heating-value, B.T.U. per pound, actual fuel	14.321
Heating-value, B.T.U. per pound, dry fuel	14.450

Per square foot of the producer fuel-bed there were gasified:

Maximum fuel	14.97 pounds per hour.
Minimum fuel	12.71 pounds per hour.
Average	13.33 pounds per hour.

The average gas-analysis during the 51-hour test was, approximately:

Carbon monoxide	23.5
Hydrogen	8.5
Methane	1
Nitrogen	63
Carbon dioxide	4
	100.0

Heating-value, by calorimeter.. 114.26 B.T.U. per cubic foot.
Heating-value, by analysis .... 112.40 B.T.U. per cubic foot.

The construction of the diagram, Fig. 178, is readily seen. Taking, for an illustration, the point *A* on the cubic feet gas per hour curve, which corresponds to a total gas-consumption of 30,000 cubic feet per hour, we obtain the corresponding heating-value consumed per hour, 3,190,000 B.T.U., as indicated on the B.T.U. per hour curve, and read on the B.T.U. per hour scale, near the right-hand side of the diagram. The horse-power corresponding to the point *A*, read on the horizontal bottom line of the diagram, is, approximately, 225 B.H.P. The heating-value consumed per brake horse-power per hour, therefore, 14,170 B.T.U., and the thermal efficiency approximately, $\frac{2,545}{14,170} = 17.9$ per cent. The latter two figures are read on, respectively, the B.T.U. per B.H.P. hour scale and the thermal efficiency scale, at the right-hand side of the diagram.

The divisions of the B.T.U. per B.H.P. hour scale and the thermal efficiency scale are such that the distance between each two successive horizontal lines represents, respectively, 800

TABLE XXXIV.

Fuel Consumption Tests on Various Loads—Holder Drop Test.

Number of Test.	A.	B.	C.	D.	E.	Remarks.
Duration of test—minutes	11	8	10	10	10	
Load rating per cent of full load	No load	25	45	70	Full load.	Circumference of holder 110.33 ft.
Speed of engine—revolutions per minute	158	156	154	152	150	
Brake horse-power	....	127.0	225.5	353.0	511.5	
Kilowatt	....	84.1	154.3	243.5	352.0	
Holder drop, ft. per hour	16.91	24.96	32.22	39.89	51.60	Average temperature of the gas 71.6° F. Barometer pressure 29 26 inches. Av. pressure of the gas $2\frac{7}{8}$ inches of water.
Gas consumption, reduced to 60° F. and 30 in. mercury:						
Total per hour, cubic feet	15760	23270	30050	37280	48200	Reduction factor 0.9642.
Per B.H.P. per hour, cubic feet	....	183.2	133.2	105.5	94.25	
Per kilowatt per hour, cubic feet	....	276.8	194.8	153.1	137.0	
Lower heating-value of the gas B.T.U. per cub.ft.	106.4	106.4	106.4	106.4	106.4	Average of all tests.
Heating-value consumed per B.H.P. per hour—B.T.U.	....	19480	14160	11215	10030	
Thermal efficiency referred to B.H.P.	....	13.05	17 96	22.68	25.36	

TABLE XXXV.

Average Heat Distribution During a 51-Hour Test (Av. 500 B.H.P.).

	Engine Only.	Entire Plant.
Useful work—B.H.P.	24.9	18.38
Friction and pump work	4.58	3.37
Loss in cooling water	34.22	25.22
Loss in exhaust and through radiation	36.3	28.81
Loss in producer		26.22
	100.00	100.00

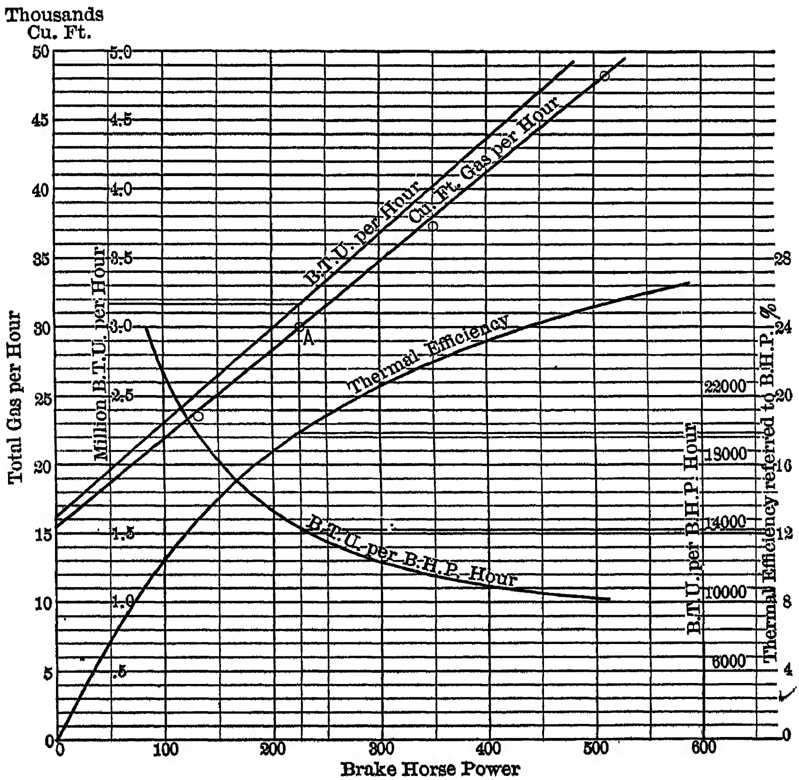


FIG. 178.—Efficiency Test of a Westinghouse 500 Horse-Power Gas-Engine.

B.T.U., and 0.8 per cent. The heating-value of the gas is figured at 106.4 B.T.U. per cubic foot.

TEST OF A NIEL SINGLE-CYLINDER 45 HORSE-POWER ENGINE on city gas. The test was made at Evreux, France, Nov. 11, 1901, by A. Witz and Moreaux.

The unit of work of the metric system is not exactly the same as that of the English; the English horse-power being approximately 1.4 per cent larger than the metric. This fact is often disregarded when data of the tests are transposed from one system to the other. To avoid discrepancies in the figures, however, due allowance should be made for the difference; or the fact that work is expressed in metric horse-power should be stated.

In the following test the data are given both in the original metric system and in the English, and, as a result of the discrepancy between the two horse-powers, it will be found that no two corresponding figures of the two columns referring to consumption per horse-power are, directly, reducible from one to the other.

Results of a Test of a Niel Four-Cycle Engine on City Gas.

	METRIC SYSTEM.	ENGLISH SYSTEM.
Diameter of the cylinder	350 $^m/_m$	13.78 in.
Length of the stroke	480 $^m/_m$	18.9 in.
A rope-brake was used, having:		
A sheave diameter	2*m*.220	84.7 in.
An effective diameter to the centre of the ropes	2*m*.254	88.74 in.
The circumference corresp. to the effective diam.	7*m*.081	278.75 in.
NO-LOAD TEST:		
Revolutions per minute	219.06	....
Gas-consumption per hour	9.640 *c.m.*	340.4 *c.f.*
Barometric pressure	763$^m/_m$	....
The pressure of the gas (by water column)	36$^m/_m$	1.42 in.
The temperature of the gas	10° C.	50° F.
Gas consumption per hour reduced to 32° F. and 760$^m/_m$.	9.331 *c.m.*	330 *c. f.*
Cooling water used per hour	255 litre	562 lbs.
Temperature of cooling water at inlet	12° C.	53.6° F.
Temperature of cooling water at outlet	67° C.	152.6° F.
Heating-value carried off by the water	14,025 *cal.*	55,640 B.T.U.

Results of a Test of a Niel Four-Cycle Engine on City Gas.—(*Continued.*)

	Metric System.	English System.
Full-Load Test:		
Duration of test	1 hour 45 m.	....
Speed of engine (rev. per minute)	213.36	....
Gas-consumption per hour	20.987 *c. m.*	741 *c. f.*
Barometric pressure	$763^{m}/_{m}$	....
The pressure of the gas (by water column)	$36^{m}/_{m}$	1.42 in.
The temperature of the gas	10.35° C.	50.63° F.
Gas-consumption per hour, reduced to 32° F. and $760^{m}/_{m}$	20.143 *c. m.*	711. *c. f.*
Cooling water used per hour	419 litre	924 lbs.
Temperature of the cooling water at inlet	12.5° C.	54.5° F.
Temperature of the cooling water at outlet	71.5° C.	160.7° F.
Heating-value carried off by the water per hour	24,721 *cal.*	98,129 B. T.U.
Net load on brake	137.083 *Kg.*	302.21 lbs.
Brake horse-power	46.02	45.38
Gas-consumption per B.H.P., gas of 32° F. and $760^{m}/_{m}$	0.439 *c. m.*	15.67 *c. f.*
Higher heating-value of the gas, at 32° F. and $760^{m}/_{m}$	5,664 *cal.* per *c. m.*	637 B.T.U. per *c. f.*
Heating-value consumed per hour per B.H.P.	2,486 *cal.*	9,981 B.T.U.
Heating-value corresponding to 1 horse-power hr.	632 *cal.*	2,545 B.T.U.
Heating-value carried off by the water, per B.H.P. per hour	537 *cal.*	2,162 B.T.U.
Indicated horse-power	53.54	52.79
Mechanical efficiency	0.859	0.859
Gas-consumption per I.H.P. per hour	0.376 *c. m.*	13.47 *c. f.*

Distribution of the heating-value in one pound of gas:

Loss in the cooling water	0.216
Loss in the exhaust gases	0.388
Loss in friction and radiation	0.141
Converted into brake horse-power	0.255
	1.000

The Prony Brake.—The brake referred to in the above engine-test consists of a brake-sheave around which is wound a rope of sufficient strength for absorbing the power of the engine. One end of the rope is fixed to a frame resting on the platform-scale

which measures the pull exerted in the rope, due to the friction between it and the brake-sheave, while the other end is tightened so as to produce the necessary frictional resistance. For the testing of smaller engines the rope-brake is a suitable appliance.

In Fig. 179 is shown a design for a Prony brake, which was originally illustrated and described by Mr. Oliver (page 1054 of the Transactions of the American Society of Mechanical Engineers, Vol. XXIV.), and which can be used successfully for absorbing much greater power than that for which the rope-brake is suitable.

The brake-wheel is cooled by means of a stream of water admitted between the flanges on the inside of the rim, and, to effect circulation of the water, a scoop is arranged which, when the wheel revolves, scoops up and carries off the hot water. The brake-shoes are built up of ribs of soft wood, poplar or basswood, with one-quarter of an inch space between the ribs, to afford access for the lubricant underneath the brake-shoe. The knife-edge at the end of the brake-lever will rest on a platform-scale suitable for measuring the torque of the engine.

In order to simplify the figures involved in the calculation of the power absorbed by the brake, the radial distance from the centre of the wheel to the knife-edge is preferably made of a dimension such that the circumference described by this radius becomes an even number of feet.

Some Commonly Required Reduction-Factors.—For reductions between the English and Metric Systems:

1 metre = 3.28083 foot = 39.37 inches.

1 foot = 0.304801 metre.

1 cubic metre = 35.315 cubic feet.

1 kilogramme = 2.2046 pounds.

1 pound = 0.4536 kilogramme.

1 kilogramme per square centimetre = 14.223 pounds per square inch.

1 pound per square inch = 0.0703 kilogramme per square centimetre.

1 kilogramme-metre = 7.23292 foot-pound.

1 foot-pound = 0.1382 kilogramme-metre.

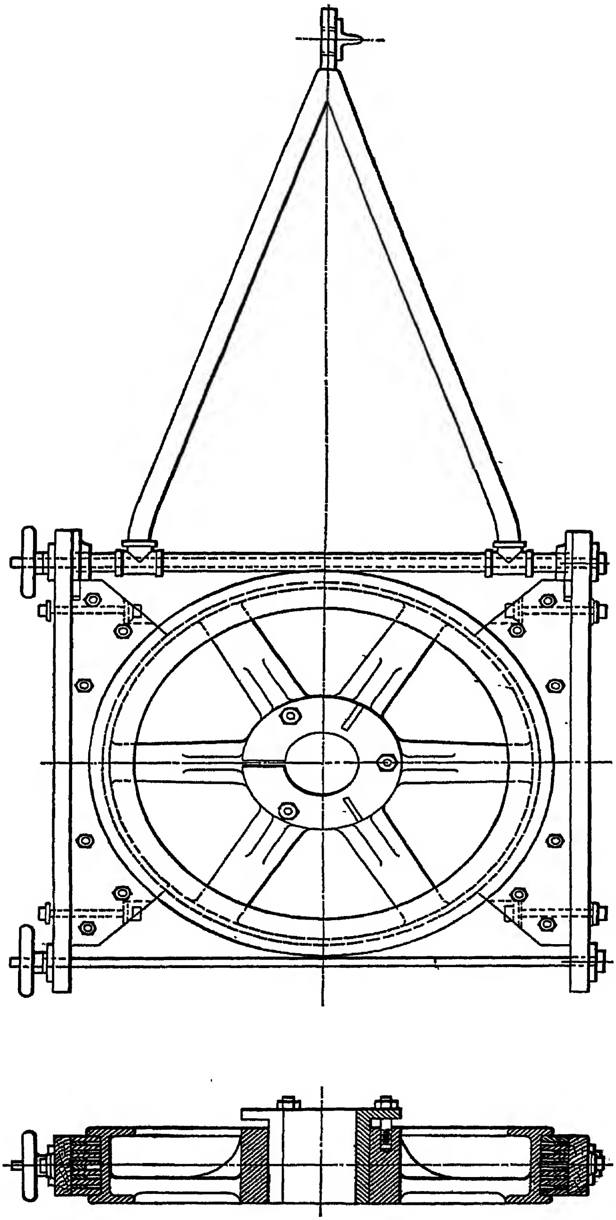


FIG. 179.—Prony Brake.

4500 kilogramme-metre per minute = 0.986 horse-power (English) = 1 horse-power (metric).

33000 foot-pound per minute = 1.014 horse-power (metric) = 1 horse-power (English).

1 calorie = 3.9683 B.T.U.

1 calorie per kilogramme = 1.8 B.T.U. per pound.

1 B.T.U. per pound = 0.555 calories per kilogramme.

1 calorie per cubic metre = 0.1124 B.T.U. per cubic foot.

1 B.T.U. per cubic foot = 8.91 calories per cubic metre.

Fuel consumption, in grammes per metric horse-power = 0.002235 pounds per English horse-power.

The mechanical equivalent of heat in the metric system is 427. Hence:

1 metric horse-power per minute = 10.54 calories per minute.

1 metric horse-power per hour = 632 calories per hour.

The Accelerating Force Due to the Reciprocating Parts.—The distance S from the cross-head to its head-end-centre position,

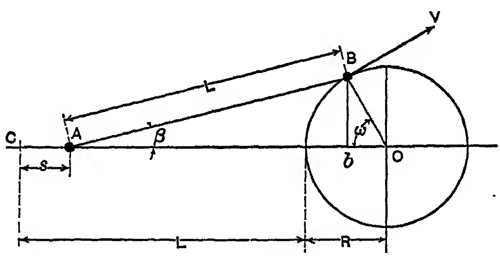


FIG. 180.

expressed in terms of the angular distance between the crank and the head-end dead centre is, according to Fig. 180

$$S = CA = CO - AO = (L + R) - (L \cos \quad + R \cos w)$$
$$= R(1 - \cos w) + L(1 - \cos \beta);$$

and we have
$$\frac{R}{L} = \frac{\sin \beta}{\sin w},$$

thus
$$\cos \beta = \sqrt{1 - \left(\frac{R}{L}\right)^2 \sin^2 w}.$$

If in the above square root we add the term $\left(\frac{R}{L}\right)^4 \sin^4 w$,

which is of inconsiderable magnitude compared with the other quantities of the root (its maximum value being $\frac{1}{256}$ when the maximum value of $\left(\frac{R}{L}\right)^2 sin^2\, w$ is $\frac{1}{16}$) then we obtain, approximately

$$cos\ \beta = \sqrt{1 - \left(\frac{R}{L}\right)^2 sin^2\, w + \left(\frac{R}{L}\right)^4 sin^4\, w}, \text{ or}$$

$$cos\ \beta = 1 - \tfrac{1}{2}\left(\frac{R}{L}\right)^2 sin^2\, w.$$

This value inserted in the expression for S gives

$$S = R\left(1 - cos\ w + \tfrac{1}{2}\frac{R}{L} sin^2\, w\right).$$

The velocity of the cross-head is

$$v = \frac{d\,s}{d\,t} = \frac{d\left[\ R\left(1 - cos\ w + \tfrac{1}{2}\frac{R}{L} sin^2\, w\right)\ \right]}{d\,t}$$

$$= R\left(sin\ w + \tfrac{1}{2}\frac{R}{L} sin\ 2\,w\right)\frac{d\,w}{d\,t};$$

when t is a variable representing time.

But as the velocity of the crank is assumed to be uniform, therefore $R\,d\,w = V\,d\,t,$

$$\text{or } \frac{d\,w}{d\,t} = \frac{V}{R},$$

and hence $v = V\left(sin\ w + \tfrac{1}{2}\frac{R}{L} sin\ 2\,w\right).$

The acceleration of the velocity is

$$a = \frac{d\,v}{d\,t} = \frac{d\left[\ V\left(sin\ w + \tfrac{1}{2}\frac{R}{L} sin\ 2\,w\right)\ \right]}{d\,t}$$

$$= V\left(cos\ w + \frac{R}{L} cos\ 2\,w\right)\frac{d\,w}{d\,t},$$

or by substituting $\frac{V}{R}$ for $\frac{d\,w}{d\,t}$ we get

$$a = \frac{V^2}{R}\left(cos\ w + \frac{R}{L} cos\ 2\,w\right).$$

The force required for giving this acceleration is

$$P = \frac{G}{g} a;$$

when G is the weight of the reciprocating parts and g the acceleration due to gravity.

Thus, for the forward stroke of the piston,

$$P_1 = \frac{G}{g} \frac{V^2}{R} \left(\cos w + \frac{R}{L} \cos 2 w \right). \quad . \quad . \quad (100f)$$

For the return stroke we obtain

$$P_2 = \frac{G}{g} \frac{V^2}{R} \left(\cos w - \frac{R}{L} \cos 2 w \right); \quad . \quad . \quad (100r)$$

if w be figured from the crank-end centre.

For the beginning of the stroke ($w = 0°$), $\cos w = 1$ and $\cos 2 w = 1$, and for the end of the stroke ($w = 180°$), $\cos w = -1$ and $\cos 2 w = 1$.

Hence we get, if r and l be expressed in inches:

The accelerating or retarding force at the head end of the piston stroke

$$P_1 = \pm \frac{12 G}{g} \frac{V^2}{r} \left(1 + \frac{r}{l} \right), \quad . \quad . \quad . \quad (100h)$$

and the accelerating or retarding force at the crank end of the piston stroke

$$P_2 = \pm \frac{12 G}{g} \frac{V^2}{r} \left(1 - \frac{r}{l} \right). \quad . \quad . \quad . \quad (100c)$$

At the point of the stroke when the connecting-rod forms with the crank a right angle, the crank-pin has momentarily a uniform and maximum velocity in the direction of the rod; the angle β is there a maximum, hence the velocity of the cross-head is, at the moment, uniform and a maximum. The velocity of the reciprocating parts, thus, changing at that point from an accelerating to a retarding one, the accelerating force is

$$P = \pm O.$$

On account of the approximation that has been introduced, this does not appear from the formulas 100 f and 100r.

The crank angle for $P = O$ is $\tan a = \frac{l}{r}$.

The Tangential Crank-Effort.—Assume, according to the notations of Fig. 181, P to be the pressure acting on the piston when the crank stands in the position $O\,B$; Q the component of this pressure in the direction of the connecting-rod, and T the corresponding tangential crank-effort, then, it will readily be seen,

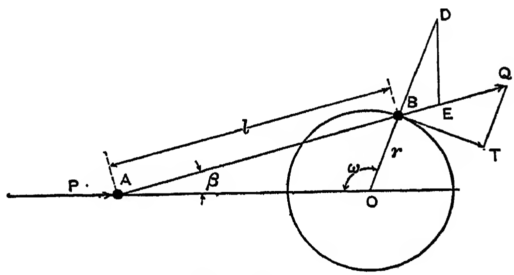


FIG. 181.

the relation between these three quantities, for any position of the crank, will be expressed by the equations

$$T = Q \sin (w \pm \beta) = \frac{P \sin (w \pm \beta)}{\cos \beta} \quad . \quad . \quad (127)$$

The sign of plus applying for all positions of the connecting-rod above the horizontal position and that of minus applying for its positions below the horizontal line.

The value of $\cos \beta$ and β will be obtained from the ratio

$$\frac{r}{l} = \frac{\sin \beta}{\sin w}.$$

Thus $$\cos \beta = \sqrt{1 - \left(\frac{r}{l}\right)^2 \sin^2 w}.$$

By inserting in equation 127 the values for P, w, β and $\cos \beta$ the tangential effort for any crank-position may be solved.

When the tangential effort for a number of crank-positions is required, it is, however, much quicker work to derive the crank-effort by means of a graphical construction as follows:

Let it be required to find the tangential crank-effort for a crank-position $O\,B$; the pressure in the cylinder for the corre-

sponding position of the piston being P. The position of the cross-head pin, A, corresponding to the position, B, of the crank-pin we find by pointing off, from B to A, the length of the connecting-rod, l; and through the points A and B we draw the line $A\,B$ representing the centre-line of the connecting-rod, which, for positions of B to the right of the vertical centre line through O, we extend a proper amount beyond the point B.

On the crank-radius extended outside of the crank-pin circle point off, from B, the length $B\,D$ representing to a suitable scale the piston pressure, P, and draw from D, perpendicularly to the base-line $A\,O$, a line $D\,E$ terminating against the centre-line $A\,B$.

We have then

$$\frac{D\,E}{D\,B} = \frac{\sin(w+\beta)}{\sin(90+\beta)},$$

or

$$D\,E = \frac{\sin(w+\beta)}{\cos\beta}\,P.$$

Hence

$$D\,E = T.$$

This construction may be carried out for any number of points of the crank-pin circle, and the required construction-lines for the various points will always be entirely clear of each other.

Engine Belts.—Engine belts being, generally, as far as the designer of an engine has control, all of identical quality, and as they work, in most all cases, under practically the same conditions, the Nagle formula (see page 878, Kent's hand-book), can for convenience with respect to such belts be written

$$w = \frac{(7 \text{ to } 9)\ B.H.P.}{c\,V}; \quad . \quad . \quad . \quad . \qquad (128)$$

w being the width of the belt, in inches,
V the rim-velocity of the belt-sheaves, in feet per second;
$B.H.P.$ the number of brake horse-power transmitted, and
c a coefficient.

If both belt-sheaves are of the same diameter then
c becomes $= 1$; if not, c should be determined according to the number of degrees arc of contact the belt forms on the smaller sheave.

The value of c will vary in the following ratio:

Arc of contact between the belt and the smaller sheave,	90°	100°	110°	120°	130°	140°	150°	160°	170°	180°
coefficient c,	0.60	.65	.70	.75	.80	.85	.90	.94	.97	1.0.

For a normal belt speed, 80 feet per second, this formula allows, when the driving and driven sheaves are of equal size, 9 to 11 horse-power per each inch width of the belt; between which limits the actual allowance in successful belt transmissions is often found to lay. The choice between the wide and narrow belt must, of course, be made according to the importance of any particular case in hand.

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